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VALVE GEARS
MECHANICS OF THE STEAM ENGINE
STEAM-ENGINE GOVERNORS
STEAM-ENGINE DESIGN
TYPES OF STEAM BOILERS
BOILER FITTINGS AND ACCESSORIES
BOILER SETTINGS AND CHIMNEYS
BOILER PIPING AND AUXILIARIES
FUELS AND BOILER TRIALS
STEAM-BOILER DESIGN

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PREFACE

The International Library of Technology is the outgrowth of a large and increasing demand that has arisen for the Reference Libraries of the International Correspondence Schools on the part of those who are not students of the Schools. As the volumes composing this Library are all printed from the same plates used in printing the Reference Libraries above mentioned, a few words are necessary regarding the scope and purpose of the instruction imparted to the students of—and the class of students taught by—these Schools, in order to afford a clear understanding of their salient and unique features.

The only requirement for admission to any of the courses offered by the International Correspondence Schools, is that the applicant shall be able to read the English language and to write it sufficiently well to make his written answers to the questions asked him intelligible. Each course is complete in itself, and no textbooks are required other than those prepared by the Schools for the particular course selected. The students themselves are from every class, trade, and profession and from every country; they are, almost without exception, busily engaged in some vocation, and can spare but little time for study, and that usually outside of their regular working hours. The information desired is such as can be immediately applied in practice, so that the student may be enabled to exchange his present vocation for a more congenial one, or to rise to a higher level in the one he now pursues. Furthermore, he wishes to obtain a good working knowledge of the subjects treated in the shortest time and in the most direct manner possible.

In meeting these requirements, we have produced a set of books that in many respects, and particularly in the general plan followed, are absolutely unique. In the majority of subjects treated the knowledge of mathematics required is limited to the simplest principles of arithmetic and mensuration, and in no case is any greater knowledge of mathematics needed than the simplest elementary principles of algebra, geometry, and trigonometry, with a thorough, practical acquaintance with the use of the logarithmic table. To effect this result, derivations of rules and formulas are omitted, but thorough and complete instructions are given regarding how, when, and under what circumstances any particular rule, formula, or process should be applied; and whenever possible one or more examples, such as would be likely to arise in actual practice—together with their solutions—are given to illustrate and explain its application.

In preparing these textbooks, it has been our constant endeavor to view the matter from the student's standpoint, and to try and anticipate everything that would cause him trouble. The utmost pains have been taken to avoid and correct any and all ambiguous expressions—both those due to faulty rhetoric and those due to insufficiency of statement or explanation. As the best way to make a statement, explanation, or description clear is to give a picture or a diagram in connection with it, illustrations have been used almost without limit. The illustrations have in all cases been adapted to the requirements of the text, and projections and sections or outline, partially shaded, or full-shaded perspectives have been used, according to which will best produce the desired results. Half-tones have been used rather sparingly, except in those cases where the general effect is desired rather than the actual details.

It is obvious that books prepared along the lines mentioned must not only be clear and concise beyond anything heretofore attempted, but they must also possess unequaled value for reference purposes. They not only give the maximum of information in a minimum space, but this information is so ingeniously arranged and correlated, and the indexes are so full and complete, that it can at once be made available to the reader. The numerous examples and explanatory remarks, together with the absence of long demonstrations and abstruse mathematical calculations, are of great assistance in helping one select the proper formula, method, or process and in teaching him how and when it should be used.

This volume comprises two distinct parts, one treating of steam-engine design and the other of steam-boiler design. In order to approach the design of a steam engine intelligently, the designer must know the manner of steam distribution, the various forces acting, and the required speed regulation. To this end, the steam-engine design papers are prefaced by descriptions of various types of valve gears and governing mechanisms, as well as a study of the forces due to inertia and centrifugal action. The design of engine parts is then taken up in logical order. The subject of steam-boiler design is presented in a similar way. First there are given a number of papers descriptive of the various types of steam boilers, their fittings, accessories, settings, piping and auxiliaries, as well as an extended discussion of fuels, combustion, and methods of making trials of boilers. These are followed by papers on design, in which are given formulas and rules for calculating the sizes and strengths of the various parts.

The method of numbering the pages, cuts, articles, etc. is such that each subject or part, when the subject is divided into two or more parts, is complete in itself; hence, in order to make the index intelligible, it was necessary to give each subject or part a number. This number is placed at the top of each page, on the headline, opposite the page number; and to distinguish it from the page number it is preceded by the printer's section mark (§). Consequently, a reference such as § 16, page 26, will be readily found by looking along the inside edges of the headlines until § 16 is found, and then through § 16 until page 26 is found.

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CONTENTS

VALVE GEARS

(PART 1)

THE SLIDE VALVE

THE PLAIN D VALVE

1. Steam Distribution.—In steam engines, the admission, cut-off, release, and compression of the steam, commonly called the **steam distribution**, are controlled by means of valves. In earlier engines, these valves took the

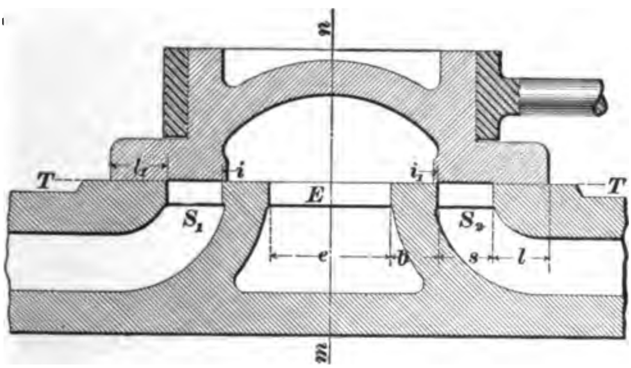


FIG. 1

form of lift valves, rising and falling upon conical seats, operated by some mechanical device. This form of valve soon gave way to the more simple **slide valve**, or common **D valve.**

Fig. 1 shows a sectional view of a plain slide or D valve. The under surface of the valve, called the **valve face,** slides over the **valve seat** *T*, *T* on the cylinder. In the cylinder are three ports. Two S_1, S_2 communicate with the passages leading to the ends of the cylinder and are called **steam ports**; the third *E* leads to the atmosphere or condenser and is termed the **exhaust port.**

In Fig. 1, the valve is in *mid-position*, with its center *n* in line with the center *m* of the exhaust port *E*.

2. The **lap** of the valve is the distance by which the edge of the valve extends beyond the edge of the steam port when the valve is in mid-position. In the case of the **D**'slide valve, the lap on the outside, as *l* and l_1, Fig. 1, is called the *outside lap*, and that on the inside, as *i* and i_1, is called the *inside lap*. Sometimes, the valve is so made as to open the port slightly to exhaust when in mid-position. A valve thus constructed is said to have **negative inside lap.**

The **lead** of the valve is the distance by which the valve has uncovered the port when the piston is at the beginning of its stroke; it is called *steam lead* when referring to the steam port, and *exhaust lead* when referring to the exhaust port. If the term lead is used without stating whether it refers to the steam or exhaust port, steam lead is always meant. The angle that the crank makes with the center line of the engine cylinder at the point of admission, when the valve just begins to open, is the angle due to the lead and is called the **lead angle.**

The **displacement** of the valve at any instant is the distance it has moved from its mid-position.

The **travel** of the valve is the total distance that the valve travels in one direction, that is, without changing its direction of motion.

The **port opening** at any instant is the distance which the valve uncovers the steam port or the exhaust port at that instant; it may be equal to the full width of the port, or it may be less.

Overtravel is the distance that the opening edge of the valve moves after the port is fully opened.

The valve is moved by an eccentric whose **throw** is equal to the diameter of the circle described by the center of the eccentric as it turns with the shaft. The radius of this circle is known as the **eccentricity.** The throw of the eccentric and the travel of the valve are the same if there is no intervening rocker-arm to increase or decrease the action of the eccentric.

The **angle of advance** of the eccentric is the angle that the center line of the eccentric makes with a line at right angles to the center line of the crank. The angle of advance is sometimes called **angular advance.** It is equal to the angle due to the lap plus the angle due to the lead.

3. Direction of Rotation.—The study of the slide valve is essentially a study of the relative motions of the piston, the crank, and the valve. The first thing to understand is the direction in which the crank will turn. This depends on the way in which the eccentric is connected to the valve and on the location of the angle of advance.

The eccentric may be connected with the valve in three ways: first, the eccentric rod may act directly on the valve; second, it may act through a rocker pivoted at one end, or one of the nature of a bell-crank; third, it may act through a reversing rocker pivoted near the middle. In the first two instances, the valve will move with the eccentric and the connection may be said to be direct. In these cases, the eccentric will always be in advance of the crank, in the direction in which the crank is to turn, by an angle equal to 90° plus the angle of advance. That is, the crank will follow the eccentric.

The action of a reversing rocker is simply to cause the valve to move in a direction opposite to that in which the eccentric is moving. Hence, when a reversing rocker is used, the eccentric will be behind the crank by an angle equal to 90° minus the angle of advance. That is, the crank will lead, and the eccentric will take a position exactly opposite to that taken in the previous case.

In some forms of valves, mostly of the piston type, steam is taken at the middle of the valve and is exhausted at the ends. When this is the case, the eccentric is set in a position diametrically opposite to that which it would occupy if the valve had been of the ordinary **D** type and connected directly to the eccentric rod. In the following discussion, the engine is assumed to take steam at the ends of the valve, and the latter is supposed to be connected directly to the eccentric rod, unless otherwise stated.

4. Displacement of the Valve.—The next thing to consider is the position of the valve for any given position of the piston. Suppose, at first, that the effects of the angularity of the connecting-rod and eccentric rod are neglected. Then, the displacement of the valve for any given piston position can be found as follows:

Draw the outer semicircle *A B C*, Fig. 2, on one side of the

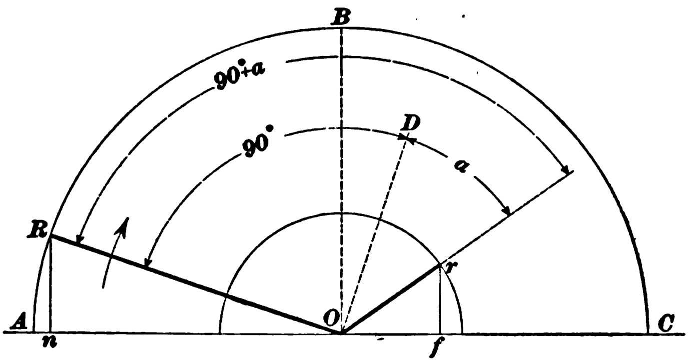


FIG. 2

stroke line *A C*, with a radius *O R* equal to the length of the main crank. This semicircle will represent the path of the crankpin *R* during one stroke. From the same center, draw the inner semicircle with a radius *O r* equal to the length of the eccentric radius. Now, suppose the piston to have moved along its stroke *A C*, a distance *A n*. The crank will then be in the position *O R*, *R* being perpendicularly above *n*. If *a* equals the angle of advance, the valve crank or eccentric will be at *O r*, ahead of the main crank by the angle *R O r*, equal to $90° + a$; and it is evident that the displacement of the valve from mid-position will then be equal to Of. By *valve crank* is meant an imaginary crank, supposed to replace the eccentric, that has a radius equal to the radius of the eccentric.

THE BILGRAM VALVE DIAGRAM

5. A more convenient method of finding the displacement of the valve is shown in Fig. 3. It forms the basis of a valve diagram known as the **Bilgram diagram,** which will be explained later, and by which a slide valve can be correctly proportioned and examples involving lead, lap, angular advance, etc. can be solved.

Let the crank and eccentric circles be drawn, as before, on one side of the stroke line AC. From O draw the line OF,

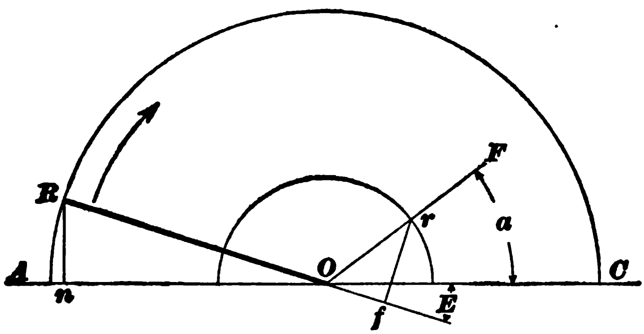


FIG. 3

making the angle a with OC equal to the angle of advance. Then, the length of a line perpendicular to the center line of the main crank, and drawn from r, the point of intersection of OF and the inner or eccentric circle, will represent the displacement of the valve. In this case, the crank position is OR, and rf, perpendicular to the center line of the crank produced beyond O, is the displacement.

This statement may be proved as follows: In Fig. 4, let ebc represent the eccentric circle; AO, the position of the crank when the piston is at the end of its stroke; Or, the eccentric radius; and bOr, the angle of advance. Let this angle be represented by a. Draw a line Or_1 from O, making an angle a with Oc and cutting the eccentric circle in r_1, as shown. Imagine, now, that the crank moves forwards through an angle q and let the new crank position be OR and the new eccentric position Or_2. Extend the line OR so as to cut the eccentric circle on the opposite side, and draw $r_1 f$ at right angles to it. Also draw rm and $r_2 p$ perpendicular to Ob, which stands at right angles to ec. Then rm and $r_2 p$ represent the displacements of the valve when the crank is in the positions OA and OR, respectively.

It is desired to prove that $r_1 f = r_2 p$. Since the angles bOr and $r_1 Oc$ both represent the angle of advance a, and

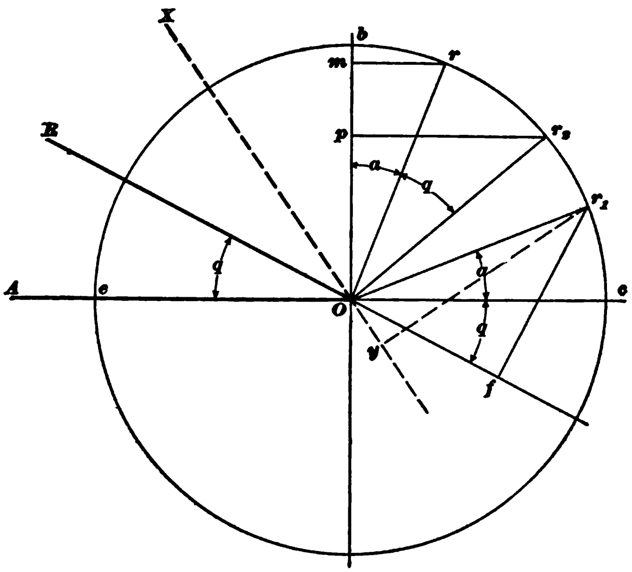


FIG. 4

the angles rOr_2 and cOf each represent the angle of motion q, the angle bOr_2, which is equal to the sum of a and q, must be equal to the angle $r_1 Of$, which is also equal to the sum of a and q. The angle $r_2 pO$ is equal to the angle $r_1 fO$, each being a right angle, and Or_2 is equal to Or_1, each being equal to the eccentric radius.

By geometry, it may be proved that when two angles and one side of one triangle are equal, respectively, to two angles and one side of another triangle, the two triangles are equal in every respect. Since in the triangles $r_2 O p$ and $r_1 O f$ two angles and one side of the one have been proved equal to two angles and the corresponding side of the other, the two triangles are equal in every respect, and $r_1 f$ must be equal to $r_2 p$. Therefore, $r_1 f$ represents the displacement of the valve when the crank is at $O R$.

When the crank stands in any other position, as $O X$, the displacement is found by dropping a perpendicular $r_1 y$ from r_1 upon $X O$ extended. The length of $r_1 y$ will then represent the displacement of the valve for that position of the crank.

6. Effect of Angularity of the Connecting-Rod. Now, suppose that the crank and the valve of an engine are moved by a connecting-rod and an eccentric, as in practice. The longer the connecting-rod, in comparison with the crank, the more nearly the motion of the piston will approach the motion of the crankpin in the same direction. Since, in most engines, the eccentric rod is very long, in comparison with the eccentricity, the relative positions of the valve and crank can be determined with sufficient accuracy by the foregoing method. The connecting-rod, however, is ordinarily only from four to six times the length of the crank. Hence, if it be required to find accurately the relative positions of the crank and piston, and hence of the valve and the piston, the effect of the angular position of the connecting-rod should be taken into consideration. In practice, this may be done as illustrated in the following example:

EXAMPLE.—Given the length of the stroke of an engine, the travel of the valve, the angle of advance, and the length of the connecting-rod; what are the valve displacements in each direction at ½ stroke of the engine?

SOLUTION.—Describe the crank-circle $A_1 R_1 C_1 R_2$, as shown in Fig. 5, with a radius equal to ½ stroke. About the same center O, describe the eccentric circle $r_1 r_2$, with a radius equal to one-half the valve travel, or the eccentricity. Draw the center line of motion $A C_1$, and on it lay off $A_1 A$, $O B$, and $C_1 C$, each equal to the length of the connecting-rod, giving the stroke $A C$ and the mid-position B of the piston. With B as a center and a radius equal to $B O$, draw arcs cutting the crank-circle at R_1 and R_2, which will give the crank positions OR_1 and OR_2, corresponding to the mid-positions of the piston. From O, draw Or_1, making an angle with OC_1 equal to the angle of advance of the eccentric and intersecting the eccentric circle at r_1. Then, the perpendicular r_1f, from r_1 to the crank position OR_1, represents the displacement of the valve for that crank position. Likewise, r_1e_1, drawn perpendicular to OR_2 extended, is the displacement for the crank position OR_2, the construction being precisely as shown in Figs. 3 and 4. The difference between r_1f and r_1e_1 indicates the difference in displacement caused by the angularity

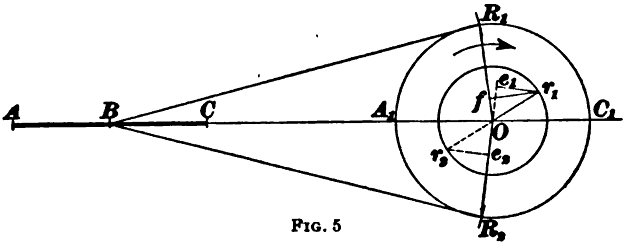


FIG. 5

of the connecting-rod when the piston is in mid-position on the two opposite strokes.

It often makes the diagram clearer to lay off the angular advance below the line of motion also, as shown at Or_2. The upper point r_1, from which the displacement is measured, is then used for crank positions from Or_2 to Or_1, and the lower point r_2 for positions from Or_1 to Or_2. In this case, the valve displacement for crank position OR_2 is r_2e_2, which, it will be seen, is equal to r_1e_1.

7. Port Opening.—Suppose the valve in Fig. 1 to move to the left. Admission of steam through the port S_2 will take place when the valve has moved a distance l equal to the lap, and the port opening will increase until the port is entirely uncovered or the valve reaches the end of its travel, when the maximum port opening will occur. This is not necessarily equal to the width of the port, as it is sometimes made less. However, the amount that the port is open at any instant, up to the point of maximum port opening, is evidently equal to the displacement from mid-position, to the left, minus the lap.

The movement of the valve to the left also opens the port S_1 to the exhaust, the amount that it is open being equal to the displacement of the valve to the left minus the inside lap i. In like manner, the openings of S_1 and S_2 to steam and exhaust, respectively, are governed by the laps l_1 and i_1 and the amount of the displacement of the valve to the right.

It must be remembered that the port opening cannot be greater than the width of the port. When a valve has overtravel, the difference between the maximum displacement and the lap will be greater than the width of the port. In such a case, the maximum displacement minus the lap gives the maximum port opening that could be obtained with a valve of the given dimensions if the port were wide enough. This would not be the actual maximum port opening. To obtain the latter, the overtravel must be deducted from the difference between the maximum displacement and the lap. The result, of course, will be the width of the port. Hence, in laying out a valve diagram for a valve with overtravel, the radius of the port-opening circle, to be described later, must be equal to the width of the port plus the overtravel.

8. Diagram for Plain Slide Valve.—Since, as has been seen, the port opening is equal to the displacement minus the lap, up to the width of the port, it can always be determined from the displacement diagram previously explained, provided that the lap is known. Moreover, as the points of admission, cut-off, compression, and release occur when the port openings to steam and exhaust are zero, the crank and piston positions for these points can easily be found.

Following are a series of valve diagrams, a sectional view of a slide valve and ports being placed under each one. Each sectional view is drawn to the scale of the diagram above it and shows the piston and valve positions corresponding to the diagram. In these diagrams, the distance the valve has moved from mid-position was found by the method already explained, and the port opening and the points of cut-off, compression, etc. were found by taking account of the laps. As a matter of convenience, the effect of the angularity of the connecting-rod has been neglected.

9. In Fig. 6, let AC represent the stroke; ABC, the crank-circle; and abc, the eccentric circle. The piston is supposed to move from left to right, and the angle FOC is the angle of advance.

When the crank position is OA, the displacement of the valve is $r_1 e_1$; the perpendicular from r_1 to OA extended; and if $r_1 l$, equal to the lap, is laid off on $r_1 e_1$, $l e_1$ will be the port opening, since the port opening equals the displacement minus the lap. As the port opening when the piston is at the end of the stroke, frequently called the dead point or dead center, is the lead, $l e_1$ is equal to the lead.

With $r_1 l$ as a radius, a circle may be described about r_1, which is called the outside-lap circle. An inside-lap circle

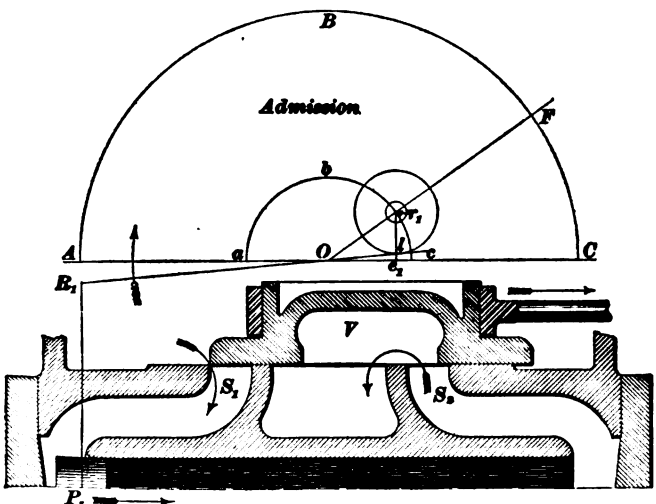


FIG. 6

may also be described about the same center with a radius equal to the inside lap. Then, if the crank is in the position OR_1, so that, when extended, its center line will be tangent to the outside-lap circle, the displacement of the valve will be equal to the outside lap, and the valve will be at the point of admission. The sectional view of Fig. 6 shows the valve V in this position, with steam just beginning to enter the cylinder through the port S_1. In the meantime, steam is being exhausted from the other end of the cylinder through the port S_2. The center of the piston is at P_1.

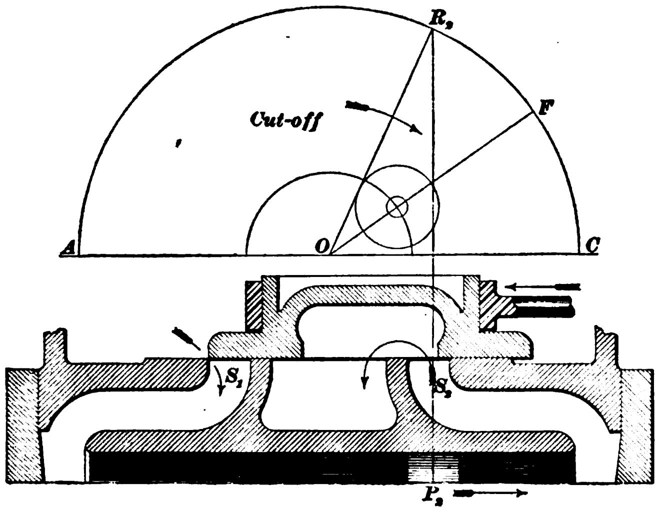


FIG. 7

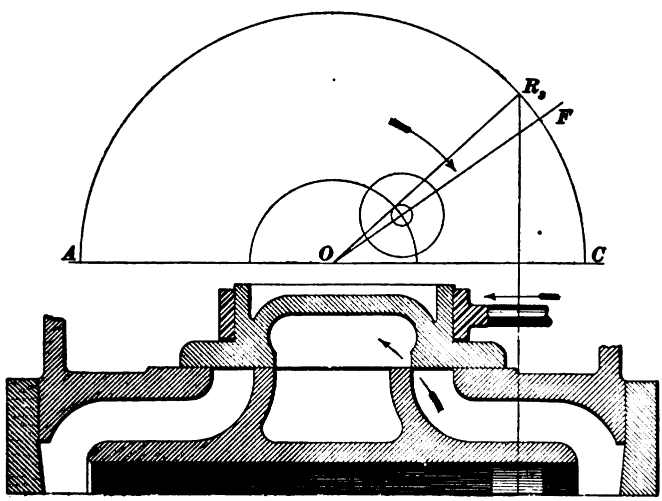


FIG. 8

10. Fig. 7 shows the crank position at OR_2, the piston being in the corresponding position at P_2. The crank, when in the position OR_2, is tangent to the lap circle, the displacement is again equal to the lap, and the steam is cut off from the left-hand end of the cylinder, but continues to exhaust from the right-hand end. The valve is in the same position as when it was just opening S_1, but now it is moving in the opposite direction and is just closing the port.

11. The next event to take place is the closing of S_2 to the exhaust at the point of compression. As the piston moves to the right and nears the end of its stroke, the crank reaches position OR_3, Fig. 8, tangent to the inside-lap circle. The displacement, therefore, is equal to the inside lap, the

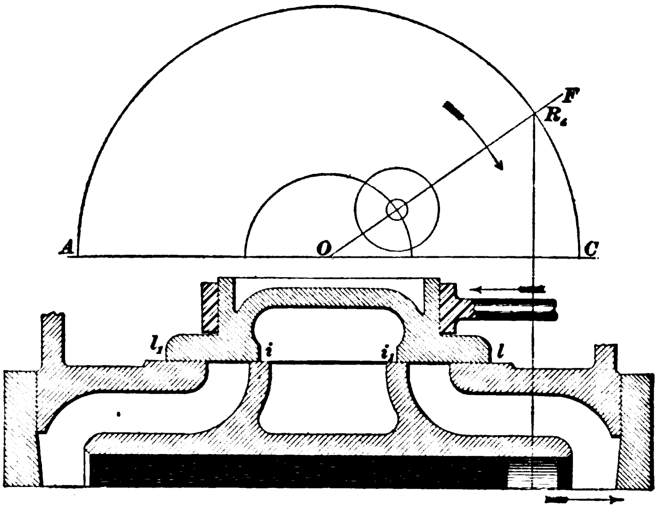


FIG. 9

valve is closed, and the steam enclosed in the right-hand end of the cylinder will be compressed during the remainder of the stroke.

12. In Fig. 9 the crank has reached the line of the angle of advance. The displacement is zero, bringing the valve in mid-position, as shown. Heretofore, the valve has been displaced to the right of the center line mn, Fig. 1, of the exhaust port, and the acting edges of the valve have been l_1 and i_1. Now, the valve is to be displaced to the left and the edges l and i are to act, so the plan before referred to of laying off the angle of advance below $A C$ is adopted.

In Fig. 10, this is done, and with r_2, the intersection of the line $O F_2$ of the angle of advance and the eccentric circle, as a center, the two lap circles are drawn corresponding to laps l and i, Fig. 9, which in this case are equal to the laps l_1 and i_1. Suppose the crank-line produced to be $O R_2$, tangent

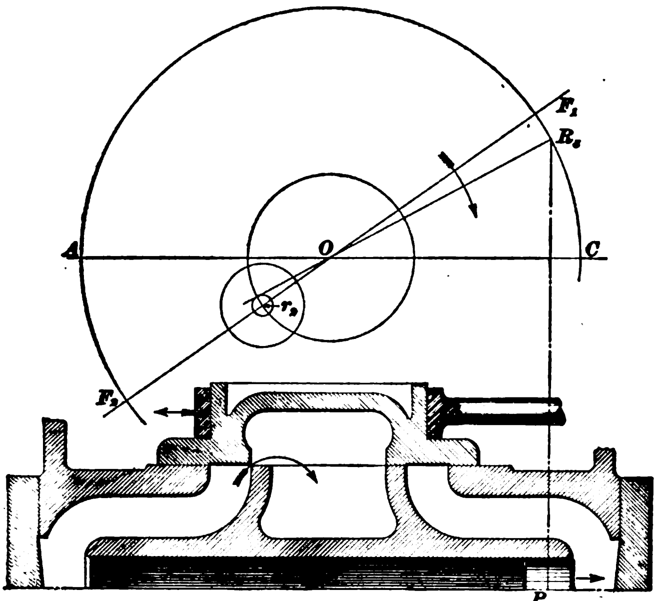


FIG. 10

to the inside-lap circle. The valve will then be at the point of release, and the steam that has been expanding in the left-hand end of the cylinder will discharge. The piston is at P_2, very near the end of the stroke, and steam will shortly be admitted to the right-hand side of the piston. Then will follow cut-off, compression, and release, as before, except that they will be for the opposite ends of the cylinder.

13. There is but one other new position to be considered, that is, maximum port opening. Fig. 11 shows the crank at OR_6, at right angles to the angle of advance line OF_6, and the piston moving to the left on the return stroke. The valve displacement is the perpendicular distance from r_6 to OR_6, which is Or_6, the greatest it can possibly be, and the port opening, which is equal to the displacement less the lap, is Ot. From the sectional views of the cylinder and valve, it will be seen that the port S_2 is wide open to take steam and the port S_1 is wide open to the exhaust. When

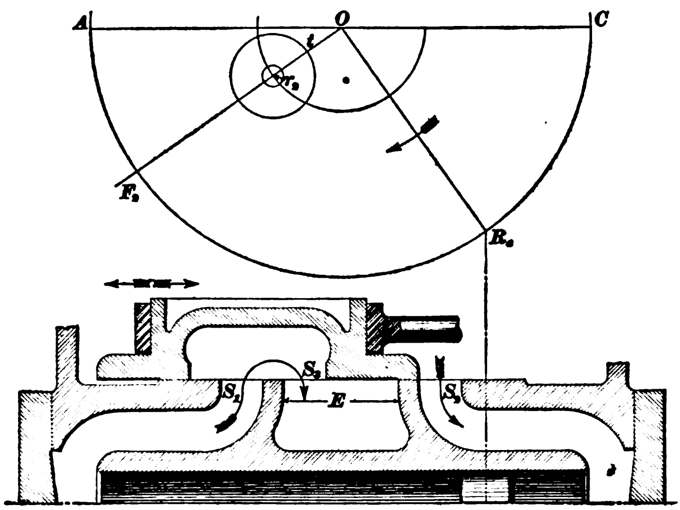


FIG. 11

the distance Ot is greater than the width of the steam port, the valve has overtravel.

14. Separate Diagram for Each End of the Cylinder.—Figs. 12 and 13 show the foregoing diagrams combined. To make them clearer, the events that take place in the left-hand end of the cylinder during one revolution are represented in Fig. 12, and those that occur in the right-hand end are represented in Fig. 13. In Fig. 12, admission begins at crank position OR_1, cut-off takes place at OR_2, release at OR_3, and compression begins at OR_4. In Fig. 13, for the other end of the cylinder, these four events occur at OR_7, OR_8, OR_{10}, and OR_2.

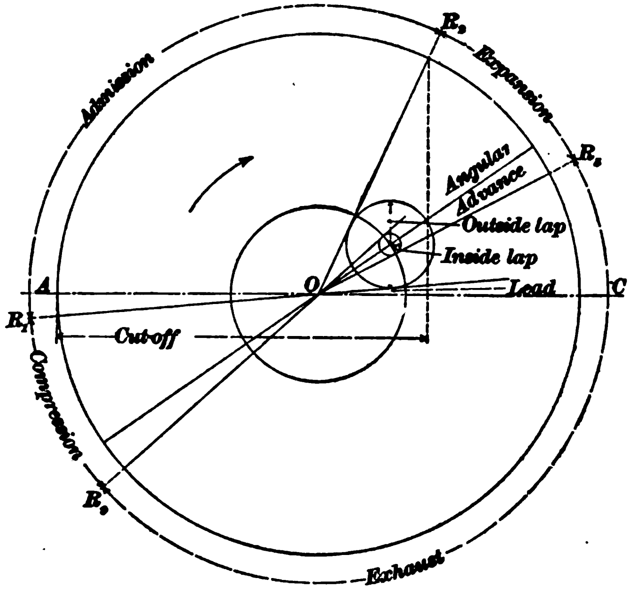

R_2
Expansion
R_3
Admission
Angular
Advance
Outside lap
Inside lap
O
A
C
Lead
R_1
Cut-off
Compression
R_4
Exhaust

FIG. 12

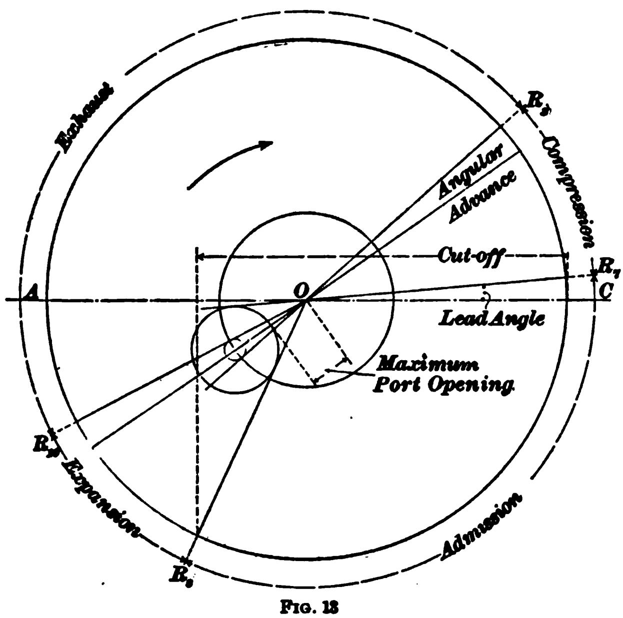

Exhaust
R_3
Compression
Angular
Advance
Cut-off
R_1
A
O
C
Lead Angle
Maximum
Port Opening
R_4
Expansion
Admission
R_2

FIG. 13

It is convenient to remember, in this connection, that as the line representing the crank position approaches the center of the lap circle, the valve is closing; and as it travels away from that center, the valve is opening. For instance, in Fig. 12, when the crank is in the position OR_2, it is approaching the lap-circle center and the valve is closing; when the crank is in position OR_1, the line OR_1 produced is moving away from the lap-circle center and the valve is opening; this is true also of the exhaust. When the crank is in position OR_6, it is moving away from the center of the lap circle, the valve is opening, and release takes place; when the crank is in position OR_8, the line OR_8 produced is moving toward the lap-circle center, the valve is closing, and compression begins.

SLIDE-VALVE PROPORTIONS

15. Areas of Ports.—The areas of the steam ports and of the exhaust ports should be large enough to avoid excessive wiredrawing, but they should not, on the other hand, be so large as to increase the clearance volume unnecessarily. In the ordinary slide-valve engine, it is customary to determine the proportions of the port for the exhaust steam, and then proportion the travel of the valve to give the required steam-port opening. The length of the port is made approximately equal to the diameter of the cylinder. It has been found, by experiment, that larger port areas are required for very high pressures than for lower pressures.

The following velocities of steam will give good results:

	Feet Per Minute
High-pressure inlet	6,500
High-pressure exhaust	5,600
Intermediate inlet	7,000
Intermediate exhaust	6,500
Low-pressure inlet	7,500
Low-pressure exhaust	7,000

The velocities in the exhaust pipe or passage from one cylinder to the next or to the condenser should be:

	Feet Per Minute
High-pressure	4,500
Intermediate	5,500
Low-pressure	6,500

16. In using the steam velocities just given, it is assumed that the entire cylinder is either filled with or emptied of steam at each stroke, and the area through the port, in square inches, multiplied by the velocity of the steam through the port, in feet per minute, must therefore be equal to the area of the cylinder, in square inches, multiplied by the piston speed, in feet per minute.

Let a_1 = area of piston, in square inches;
a_2 = area of port, in square inches;
s_1 = piston speed, in feet per minute;
s_2 = velocity of steam through port, in feet per minute.

Then $$a_2 s_2 = a_1 s_1 \qquad (1)$$

and $$a_2 = \frac{a_1 s_1}{s_2} \qquad (2)$$

EXAMPLE.—The diameter of a high-pressure cylinder is 24 inches; piston speed is 640 feet per minute; length of exhaust port is 23 inches; what should be the width of the exhaust port?

SOLUTION.—With a mean high-pressure exhaust flow of 5,600 ft. per min., the area of the port, in square inches, will be, by formula **2**, $a_2 = \frac{.7854 \times 24^2 \times 640}{5,600} = 51.7$ sq. in., nearly. This area divided by 23, the length of the port, in inches, will give the width of the port. Thus, $51.7 \div 23 = 2\frac{1}{4}$ in., nearly, is the width of the port. A similar calculation will give the area and width of the steam ports.

17. The width of the exhaust port should be such that when the valve is in its extreme position there is no reduction of its area below that of the steam port. This width may be found by the following rule:

Rule.—*To find the width of the exhaust port, add together the width of the steam port, half the travel of the valve, and the inside lap; from their sum, subtract the width of the bridge.*

18. Width of Bridge.—The **bridge** is the wall between the steam port and the exhaust port. Having determined the dimensions of the ports in the cylinder, the next step is to fix on the best width of bridge. This is generally made equal to the thickness of the cylinder and should not be much less. It should be of sufficient width to give the valve a good bearing surface, and thus prevent leakage from the steam port into the exhaust port when the valve is in its extreme positions.

19. Point of Cut-Off.—The point of cut-off must be determined next, for on this depends the outside lap and the travel of the valve. With a plain **D** slide valve, the cut-off is seldom earlier than $\frac{5}{8}$ or $\frac{3}{4}$ stroke. A high ratio of expansion requires an early cut-off, and early cut-off demands increased lap, which in turn necessitates longer travel, a larger valve, and a larger exhaust port. This increases the friction between the valve and its seat, and hence requires a larger amount of power to move the valve and causes increased wear and leakage. An early cut-off in a plain slide-valve engine also necessitates an early release and consequently an excessive compression. For these reasons, the plain slide valve is commonly used only where a cut-off as late as $\frac{5}{8}$ or $\frac{3}{4}$ stroke is not objectionable.

20. Amount of Lead.—The amount of lead given to a valve is generally decided arbitrarily and depends on the working conditions and the experience of the designing engineer; it varies from $\frac{1}{32}$ inch in small engines to $1\frac{1}{4}$ inches in the large low-pressure cylinders of multiple-cylinder engines. The piston speed and inertia of the reciprocating parts must be taken into account in determining the lead, for quite frequently it is impossible to get sufficient compression in large low-pressure cylinders to bring the reciprocating parts to rest without shock. In such cases, large, even excessive, lead must be resorted to in order to prevent such shocks. In vertical engines, the lead at the upper end should be much less than at the lower end, in order to bring the moving parts to rest without shock at the lower end of the stroke.

Generally, the lead at the lower end should be about twice that at the upper end.

21. General Problem.—Given, the stroke of an engine, 24 inches; length of connecting-rod, 45 inches; cut-off, $\frac{2}{3}$ stroke; release, $\frac{11}{12}$ stroke; lead, $\frac{1}{8}$ inch; width of steam ports, $1\frac{1}{8}$ inches; maximum port opening, 1 inch; to find the inside and outside laps, the travel of the valve, the angle of advance, and to draw a section of the ports and valve.

At first, it will be convenient to neglect the effect of the connecting-rod angularity. In laying out a diagram, it is

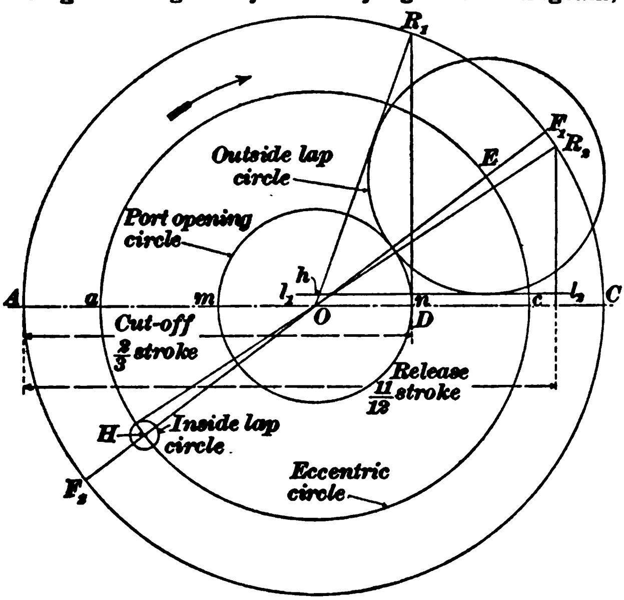


FIG. 14

not usually convenient to draw the crank-circle full size, and sometimes the eccentric circle also is drawn to a reduced scale. In this case, the crank-circle will be drawn to a scale of $1\frac{1}{2}$ inches = 1 foot, and the valve circle half size. All the measurements and parts of the diagram pertaining to the valve will, of course, be half size also.

22. From the center O, Fig. 14, on the stroke line, describe the crank-circle $A\,C$. It is required to find the outside lap when the point of cut-off, the maximum port opening, and the lead are given. Describe the circle *m n* about O, with a radius equal to the maximum port opening; draw an indefinite line $l_1 l_2$ parallel to $A C$ and above it a distance equal to the lead; finally, draw the crank position $O R_1$ for cut-off at $\frac{2}{3}$ stroke. R_1 is perpendicularly above D, and $A D$ is laid off equal to two-thirds of $A C$. D falls on the port-opening circle in this instance, but does not generally do so. By referring to Figs. 6, 7, and 11 and the accompanying descriptions, it will be evident that a circle drawn tangent to $O R_1$, $l_1 l_2$, and the circle *m n* will be the outside-lap circle, the radius of which will equal the outside lap of the valve. The center of this circle is found to be E, which point can be readily located by bisecting the angle $R_1 h l_2$; the center must then fall at some point on the bisector, and that point can easily be determined by trial.

23. To determine the valve travel, it is necessary simply to draw the eccentric circle *a c* about the center O, through the point E. The diameter of this circle will be the valve travel. The angle of advance is obtained by drawing the line $F_1 F_2$ through the points E and O, making the angle $F_1 O C$ with $A C$. This angle, $F_1 O C$, is the angle of advance.

24. Finally, to obtain the inside lap, draw the crank position $O R_2$ for $\frac{11}{12}$ stroke. Since $O R_2$ is beyond the angle-of-advance line, $O F_1$, it should properly be produced beyond O. As the laps on both ends of the valve are equal, however, this is not necessary. The radius of the circle with the center H, and drawn tangent to $O R_2$ produced, will then represent the inside lap. On measuring the diagram, the following approximate dimensions are obtained: travel, $4\frac{7}{16}$ inches; outside lap, $1\frac{7}{32}$ inches; inside lap, $\frac{1}{8}$ inch; angle of advance, 37°.

A section of the valve and ports, drawn to a scale of 3 inches = 1 foot, is shown in Fig. 1. To draw the valve, the length of the valve face must be known, and this is easily determined by first drawing a section of the ports. By the rule already given, the width of the exhaust port is $1\frac{1}{8} + 2\frac{7}{32} + \frac{1}{8} - 1 = 2\frac{15}{32}$ inches. It is drawn $2\frac{1}{2}$ inches wide, $1\frac{1}{4}$ inches on each side of the center line m, n, Fig. 1. The bridges are drawn 1 inch thick on the assumption that the cylinder walls are of that thickness, and the steam ports are then laid off beyond the outer edges of the bridges. Now, having completed the section of the ports, the outside laps should be laid off outwards from the outside edges of the steam ports, and the inside laps from the inside edges. The valve section may then be completed.

25. Equalizing the Cut-Off.—The valve that has been described is designed to cut off at $\frac{5}{8}$ stroke and to have a lead of $\frac{1}{8}$ inch, neglecting the irregularity produced by the angularity of the connecting-rod. Should an indicator be applied to such an engine, however, it would show that the cut-off occurred later than $\frac{5}{8}$ stroke on the forward stroke, and earlier on the return. In other words, steam would be admitted to the head end for a longer time than to the crank end. One way to overcome this is to give more lap to the end of the valve toward the head end of the cylinder, which will hasten its action during the forward stroke, and to reduce the lap on the other end in order to retard the action on the return stroke.

26. Fig. 15 shows a diagram for the valve laid out in this way, in which $O R_1$ and $O R_2$ are the crank positions at cut-off, and H is the center of the small outside-lap circle. The circle whose diameter is $m n$ is the port-opening circle for the head end. In this case, it happens that $m n$ is equal to one-third of the valve travel, so that the arcs through R_2 and R_1 meet $A C$ at m and n. This is not true in all cases, however. The points R_1 and R_2 are determined by laying off the distance $A n$ on the line $A C$, equal to $\frac{5}{8}$ $A C$. Then produce $A C$ to the left a distance equal to or greater than the length of the connecting-rod. With a radius equal to the length of the connecting-rod, to the same scale as that to which the crankpin circle was drawn, and with a center on $A C$ produced, describe the arc $n R_1$, intersecting the crankpin circle at R_1, which is the position of the crankpin when the steam is cut off from the head end of the cylinder. Also, lay off Cm equal to $\frac{2}{3}$ AC, and with the same radius and a center on the line AC produced, describe the arc mR_2, intersecting the crankpin circle at R_2, which is the position of the crankpin when the steam is cut off from the crank end of the cylinder.

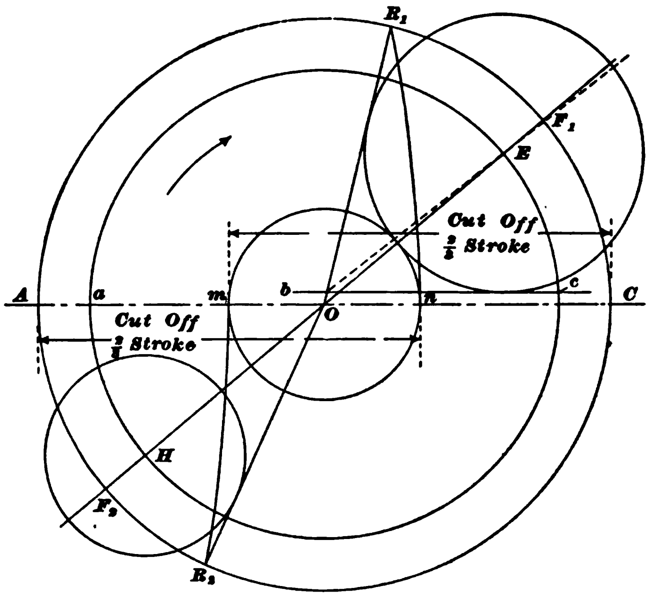


FIG. 15

Draw the lead line bc parallel to and $\frac{1}{8}$ inch above AC. Then, bisecting the angle included between OR_1 and bc and drawing the lap circle tangent to OR_1, bc and the circle mn, the center E is obtained, and OE is drawn through O and E. The eccentric circle ac is then drawn with O as a center and with OE as its radius. Extend EO until it intersects the eccentric circle in H. With H as a center draw a circle tangent to OR_2. The radius of this circle represents the outside lap on the other end of the valve.

From the preceding, it will be seen that equalizing the cut-off by varying the laps causes excessive lead, in this case nearly $\frac{1}{8}$ inch, for the return stroke. It also causes an unequal port opening, which, however, is of minor importance, provided that the opening on each end is sufficient.

27. A better way of equalizing the cut-off is to place a rocker-arm between the valve and the eccentric, which, if properly designed, will correct the irregularities due to the connecting-rod, at the points of cut-off, without disturbing the leads. The method of procedure is as follows:

First, determine from the valve diagram, in which the angularity of the connecting-rod is taken into consideration, the valve travel, lap, angle of advance, etc. for one end. Next, lay out a diagram like Fig. 16, in which *A C* represents the stroke of the crosshead, *D F B H* is the crankpin circle, and *h d f b* is the eccentric circle, with a diameter in this case smaller than the travel of the valve, because of the multiplying

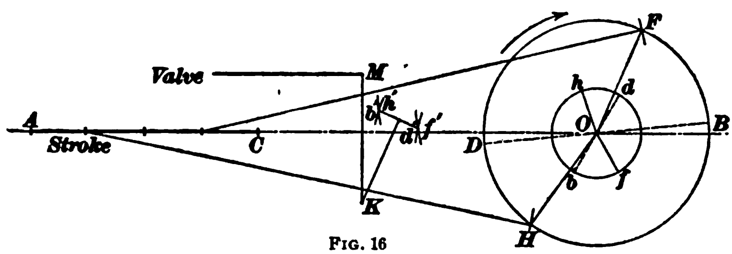


Fig. 16

effect of the rocker. The cut-off is to take place at $\frac{3}{4}$ stroke. The corresponding crank positions are *O F* and *O H*, and the crank positions at admission will be *O D* and *O B*.

When a direct rocker is used, the eccentric must be 90° plus the angle of advance ahead of the crank. In Fig. 16, the eccentric positions *O f* and *O h* are laid off 90° plus the angle of advance ahead of crank positions *O F* and *O H*, found from the diagram. In like manner, eccentric positions *O d* and *O b* are drawn corresponding to the crank position at the points of admission.

It is well known that a slide valve must be in the same position at both cut-off and admission. Hence, with a radius equal to the length of the eccentric rod, and with admission and cut-off points *d* and *f* as centers, strike arcs *d'* and *f'*.

The point of their intersection will be the point at which the eccentric-rod pin should be at admission and cut-off on the forward stroke. In like manner, the intersection of arcs b' and h' drawn from b and h as centers gives the point for the return stroke. Connecting these points and drawing a line perpendicular to the connecting line half way between them, the central position of one arm of the rocker is located. The other arm KM should be perpendicular to the valve stem at the central position, and the point of intersection K must be so chosen that the two lever arms will be proportional to the travel of the valve and the length of the line $b'd'$. By methods similar to the foregoing, the release or compression may be equalized.

Whenever, for any purpose, a rocker is used that either increases or diminishes the motion of the valve as compared with that of the eccentric, the travel of the valve must be used instead of the throw of the eccentric for all calculations and constructions connected with the valve diagram, relative to the lap, lead, or cut-off.

OTHER FORMS OF SLIDE VALVES

28. Double-Ported Valves.—Double- or multiple-ported valves are used in large engines in order to secure a wide and quick port opening with a small valve travel, thus reducing the wear and also the power required to drive them. A good example of double-ported valve, known as the **Trick valve**, is shown in Fig. 17.

In Fig. 17 (*a*), the valve is shown in mid-position. It is made with a passage h running through it; otherwise it corresponds very closely to the ordinary **D** valve already described. A movement to the right a distance equal to m will bring the edge p of the valve to the edge of the port s, as shown in Fig. 17 (*b*), so that any further movement to the right will admit steam to the cylinder. But this same movement will bring the edge f of the passage in line with the edge g, and any further movement to the right will admit steam to the passage h and, hence, to the left-hand port s.

Suppose the valve to move, say, $\frac{1}{16}$ inch to the right from the position shown in Fig. 17 (*b*), then the edge *f* will be $\frac{1}{16}$ inch beyond the edge *g*, and edge *p* will be $\frac{1}{16}$ inch beyond the outer edge of the port *s*. Thus, a movement of the valve that would ordinarily have opened the port $\frac{1}{16}$ inch, had a **D** valve been used, has opened the port twice $\frac{1}{16}$ inch, or $\frac{1}{8}$ inch. In Fig. 17 (*c*), the valve is shown in its extreme

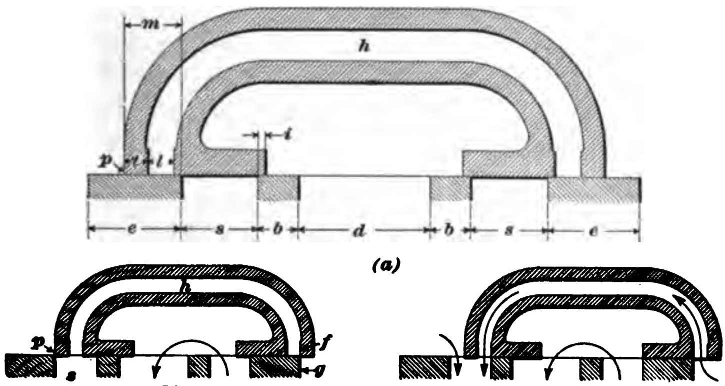


FIG. 17

right-hand position, giving full port opening to the steam and exhaust. The inside lap is shown by *i* and the outside lap is equal to *m*, Fig. 17 (*a*).

29. The parts of the valve should have the following dimensions: $l = \frac{1}{2}(s - t)$; half travel $= s + i$; $e = 2m - t$. The width of the exhaust port *d* should be equal to $s + m + i + l - b$. The diagram is drawn as for a simple valve, remembering that the width of the opening to exhaust is *s* and of the opening to steam is $2\,l$.

30. In marine engines having a large diameter of cylinder and short stroke, double-ported valves of the form shown in Fig. 18 are often used to obtain a sufficient port opening with a small travel. The steam is admitted to the cylinder at the outer edges *a*, *a* and through the passages *b*, *b*, Fig. 18 (*a*), and is exhausted through the passages *e*, *e* and at the inside edges *f*, *f*. This valve really consists of two plain D valves cast in one piece, and the plain slide-valve diagram may be applied, the port openings for each valve considered separately being one-half of the total port opening required. In Fig. 18 (*b*) is shown a half section through the exhaust passage at *g h*; and in Fig. 18 (*c*), a half section through the steam and exhaust passages at *k l*. Fig. 18 (*a*) shows a section through *m n*, Fig. 18 (*c*).

31. Piston Valves.—**Piston valves** are used principally when very high steam pressures are employed; they are, however, frequently used in connection with shaft governors where the controlling force of the governor must overcome the resistance of the valve. Piston valves are as

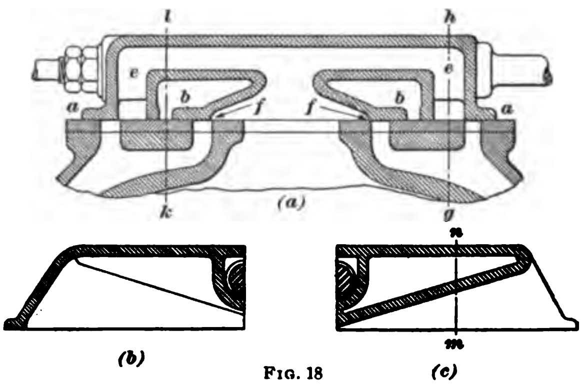


Fig. 18

nearly balanced as any type of valve yet devised, but they are liable to give trouble on account of the leakage of steam, which seems unavoidable. Piston valves moving vertically are much less subject to wear and leakage than piston valves moving horizontally, from the fact that in the vertical engine the weight of the valve does not bear on the valve seat, but on the eccentric. Many types of packing have been devised for piston valves, all of which consist of combinations of snap rings similar in principle to common piston packing rings; these lessen the leakage but add to the force required to move the valve. Owing to the wear and consequent leakage, it is not considered good practice at present to use piston valves on horizontal engines, except in cases of very high pressure.

Fig. 19 shows a modern form of piston valve used in vertical marine engines. The piston valve *a* is of the double-end type and is fitted with packing rings at *b*, *b*. The two ends work in a pair of cylindrical valve seats *c*, *c*, which may easily be removed for the purpose of renewal or repair. The valve shown is also fitted with a balance piston *d* commonly used on large vertical engine valves of the common slide-valve and balanced type, as well as on piston valves. This piston is so proportioned that the pressure acting on the lower side will balance the weight of the piston and valve.

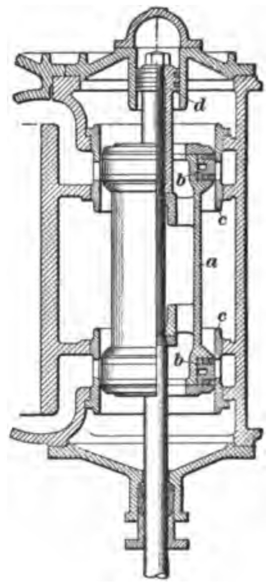


FIG. 19

32. Balanced Valves.—The advent of higher steam pressures is almost wholly responsible for the existence of the **balanced valve.** Aside from the piston valve, this balancing is accomplished in one of two ways; by providing a heavy unyielding plate over the back of the valve, as shown in Fig. 20, or by means of relief frames, as shown in Fig. 21.

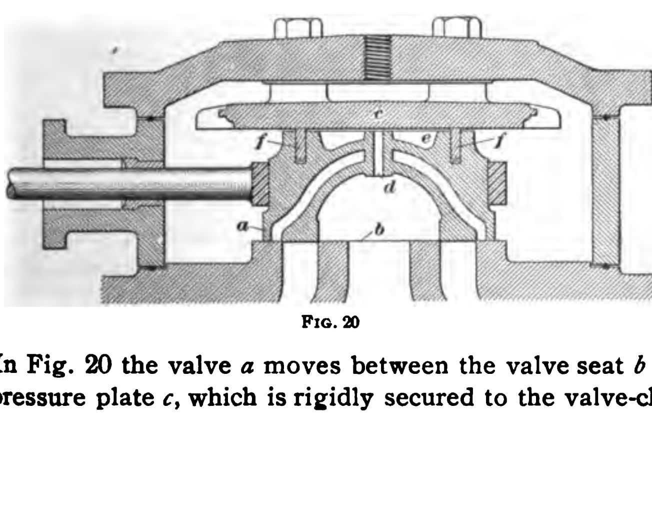


FIG. 20

In Fig. 20 the valve *a* moves between the valve seat *b* and a pressure plate *c*, which is rigidly secured to the valve-chest cover. An opening *d* through the valve causes the chamber *e* to be subjected to the exhaust pressure. The packing strips *f*, *f*, which are held against the pressure plate by means of springs, make a tight joint that readily adjusts itself so as to take up the wear.

In Fig. 21, the valve is made in two parts; the lower part *a*, which forms the valve proper, rests upon the valve seat, and the upper part *b*, which bears against the valve-chest cover and makes a telescope joint with the lower part, as shown. Packing rings *c* make the joint steam-tight. The pressure of the steam against the flange *d* holds the part *b* against the cover.

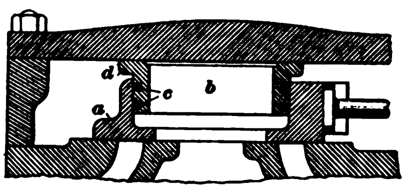


FIG. 21

In both of these types, the valve is relieved of a pressure on its seat equal to the product of the area of the relief or packing ring, and the difference between the pressure in the steam chest and that in the exhaust passage, which may be the condenser pressure.

Another form of balanced valve, and one much used in British rolling-mill practice, is shown in Fig. 22, in which *a* is the valve and *b* is a piston secured to its back and forming a working fit in the cylinder *c*, which slides upon a planed facing on the steam-chest cover. The balance piston on the back of the main valve is provided with three packing rings *d*. The chamber inside of the piston *b* is connected with the atmosphere by means of the opening *e*.

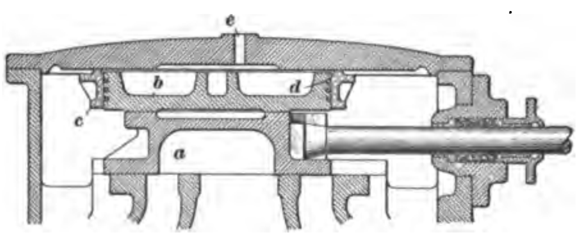


FIG. 22

THE HARMONIC VALVE DIAGRAM

33. Having determined, in a general way, by the methods already given, the elements of a valve motion, that is, the width of port, inside lap, outside lap, valve travel, etc., there still remain other points that must be settled in the designer's mind. These are very readily and accurately considered by means of a valve diagram known as the **harmonic diagram.** Some of the points that may be investigated advantageously with this diagram are the influence of the angularity of the connecting-rod, the effect of overtravel, the amount of port opening at any instant, the equalization of cut-off and its effect upon the lead, and the adjustment of release and compression.

34. Like other valve diagrams, the harmonic diagram is intended as a means whereby various valve dimensions may be determined and the valve functions closely examined. But it possesses an advantage over other forms of valve diagrams in that it shows at a glance the positions of the crank, the piston, and the valve for any angular position of the crank, or for any position of the piston in its stroke.

THE SIMPLE HARMONIC DIAGRAM

35. A harmonic valve diagram is shown in Fig. 23. The piston P is at the beginning of its forward stroke, and the valve V is therefore displaced from its mid-position an amount equal to the lap plus the lead. The crank-circle is represented by ABC, and the eccentric circle by abc. For convenience, the angularity of the connecting-rod is neglected in this illustration. First, two horizontal lines, DH and KL, are drawn from the ends of the stroke, A and C, respectively. Then the distance between these two parallel lines represents the length of the stroke. Midway between them, and parallel to both, the line xy is drawn. This line, if produced, must pass through O and hence through the center lines of the piston and valve when they are in mid-position. The line xy, therefore, represents mid-positions

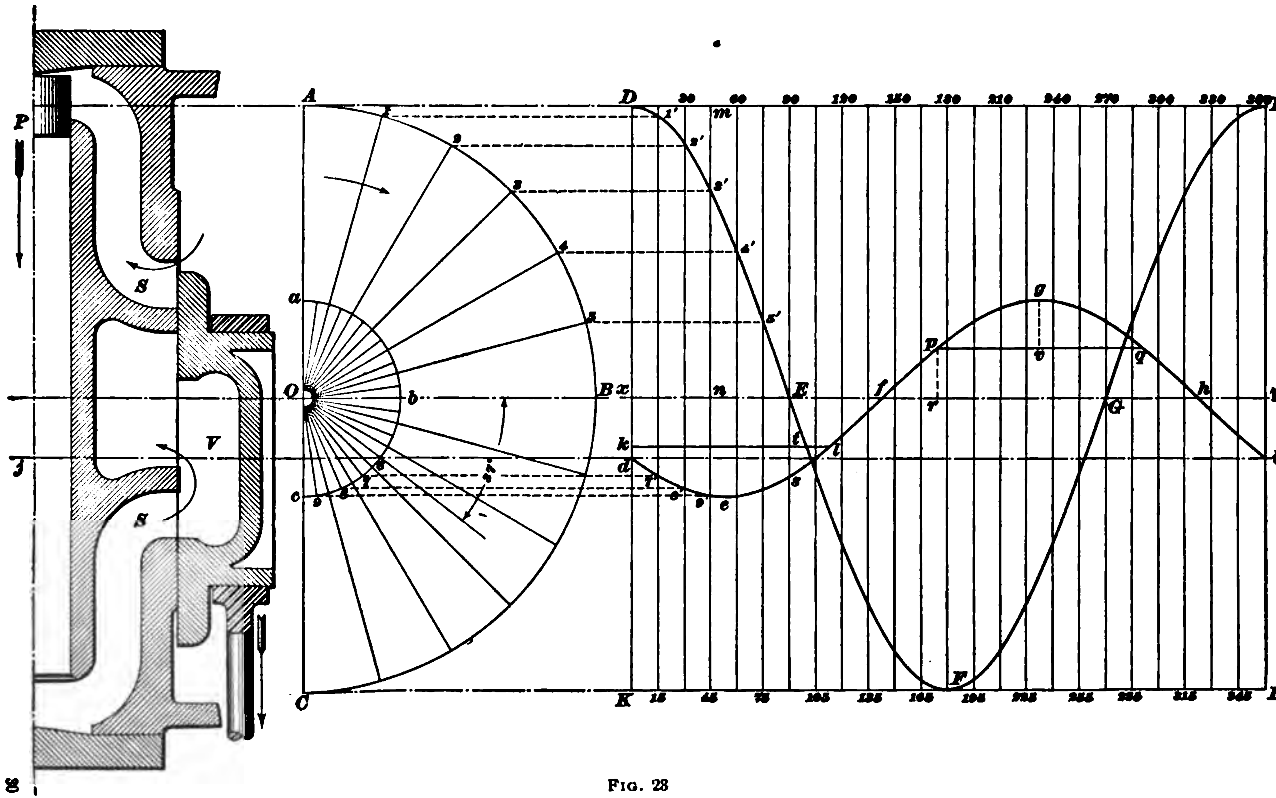


Fig. 28

of both the piston and the valve. Next, the semicircle *ABC* is divided into a number of equal arcs, in this case twelve, so that each arc represents $180 \div 12 = 15°$ of angular movement of the crank, since there are 180° in the semicircle representing half a revolution. Consequently, in a complete circle, representing an entire revolution, there will be twenty-four equal arcs, indicating that the crank passes through that number of equal angles of 15°, or 360° in all, in one revolution. The points of division on *ABC* are marked *1, 2, 3, 4, 5*, etc. Now the line *xy* is divided into twenty-four equal parts of any convenient length, corresponding to the number of 15° divisions in the complete crank-circle; and through the points of division on *xy* perpendiculars are drawn, and are continued until they meet *DH* and *KL*. Then, each of the equal spaces on *xy* represents 15° of movement of the crank, the line *DK* being the zero, or dead-center position, the next vertical line being the position of the crank 15° from dead center, the second being the 30° position; and so on for an entire revolution.

36. When the crank is at *OA*, the piston is at the end of the stroke, and a horizontal line drawn from the center of the piston, as shown, will cut *DK* at *D*. The point *D* represents the piston position when the crank is on dead center. Now, suppose the crank to move to *O1*, or 15° from *OA*. Draw a horizontal line from *1* until it cuts the first vertical line to the right of *DK* at *1'*. Then *1'* indicates the positions of crank and piston, for the point *1'* lies on the 15° line, showing that the crank has turned 15° from dead center, and the vertical distance from *1'* to the line *DH* shows the actual distance the piston has moved from the end of its stroke. Similarly, the points *2', 3', 4', 5'*, etc. are located. When the crank reaches *OB*, the piston is in mid-position, and hence the positions of crank and piston are represented by the point *E* where *xy* intersects the 90° line. When the piston reaches the end of its forward stroke, the crank has turned through 180° and is at *OC*. Hence, the point *F*, where *KL* cuts the 180° line, represents at that instant the crank and piston positions.

For the return stroke, the same method is followed, the points being plotted in reverse direction and on the vertical lines between 180 and 360. Finally, a smooth line is traced through the series of points thus located, resulting in the curve $DEFGH$, which is termed the *piston-displacement curve*, since the distance of each point on it from DH represents the distance of the piston from the end of the stroke at a particular crank position, which is found by projecting this point vertically on the line xy.

37. When the crank is at OA, the eccentric is in the position $O6$, so that the angle $bO6$ is equal to 37°, the angle of advance. If a horizontal line be drawn through *6*, it will cut DK and HL at d and i, respectively. The distance xd, then, shows the displacement of the valve from mid-position when the crank is on dead center, as does also yi, since the crank is in the same position at the beginning of à revolution as at the end. Next, from the point *6* on abc, lay off equal arcs representing 15° of angular movement of the eccentric, since the eccentric must move through the same angles as the crank. The semicircle abc will thus have a number of points, as *6*, *7*, *8*, *9*, etc. As DK represents the position of the eccentric $O6$, the point *7* must be projected horizontally to *7′* on the 15° line, the point *8* to *8′* on the 30° line, and so on. The resulting curve $defghi$ is the *valve-displacement curve*, since each point on this curve shows, by its distance from xy, the amount the valve is displaced from mid-position at the corresponding positions of crank and piston. Thus, when the crank has moved through 45°, the piston has moved a distance equal to $3'm$ and the valve is displaced from mid-position an amount equal to $9'n$. The diagram $DHLK$ is then the harmonic valve diagram.

It is a good plan, in constructing harmonic diagrams, to draw the eccentric circle full size, where possible, and to make the crank-circle to a reduced scale, but slightly larger than the eccentric circle.

38. Fig. 23 also indicates how the lap is represented in the harmonic valve diagram. At a distance $x k$ below $x y$ equal to the lap of the valve at the head end, a line $k l$ is drawn parallel to $x y$. Similarly, $p q$ is drawn parallel to $x y$ but above it a distance $p r$ equal to the lap at the crank end.

Since the port opening is equal to the displacement minus the lap, it is evident that the distance between the lap line and the valve-displacement curve at any point is the port opening at that point. For example, take the piston at the beginning of its forward stroke. The displacement of the valve at that time is $x d$ and the lap is $x k$. The port opening must, therefore, be $x d - x k = k d$, which, it will be seen, is just equal to the lead of the valve, or the amount the port is open when the crank is on dead center.

When the crank has moved through 90° and the piston is at the middle of its stroke, as represented by the point E, the displacement of the valve is $E s$, while the port opening for that crank position is $E s - E t = t s$. As the valve has passed its point of maximum displacement e, the port is evidently closing. This is further shown by the fact that the valve-displacement curve from e to f is drawing nearer to $x y$. At l, the displacement of the valve is just equal to the lap, and hence the port is just closed, while at f the displacement is zero and the valve is in its mid-position.

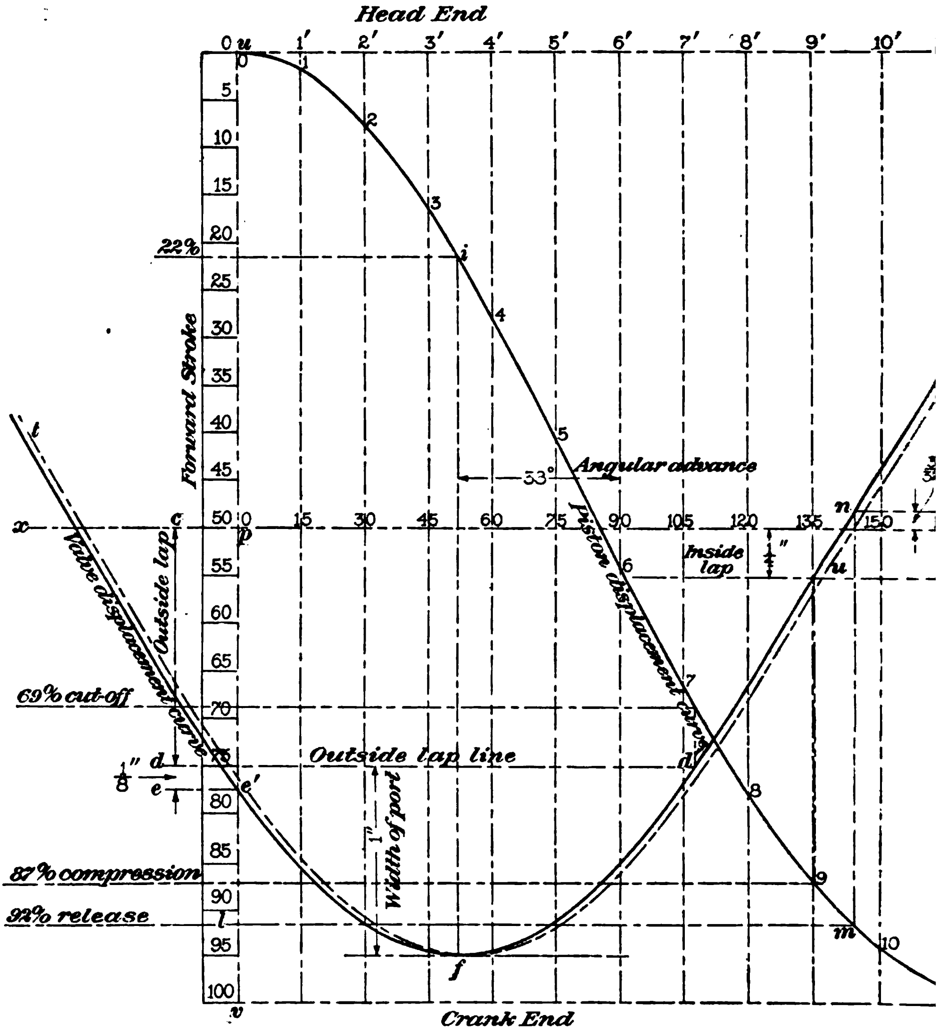
The continuation of the valve-displacement curve above $x y$ indicates that the valve is moving away from its mid-position toward the head end. At the point p, the displacement $p r$ equals the lap $p r$, and the crank-end steam port is therefore just on the point of opening for admission at that end. By the time the valve reaches g, the point of maximum displacement toward the head end, the port opening at the crank end is a maximum, and is equal to $g v$. At q, the displacement is again equal to the lap, and this is, therefore, the point at which cut-off occurs at the crank end. Hence, to find the port opening for any position of the piston or of the crank, measure the distance between the lap line and the valve-displacement curve at that position.

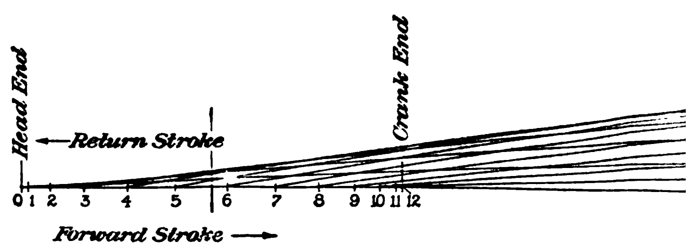
Each of the curves *DEFGH* and *defghi*, Fig. 23, represents, graphically, the movement of a point having a simple harmonic motion, as the angularities of the connecting-rod and of the eccentric rod have been neglected. If the effect of angularity is taken into account, the curve will no longer represent a true harmonic motion. However, since its shape will vary very little from that shown in Fig. 23, it is also known as the harmonic valve diagram.

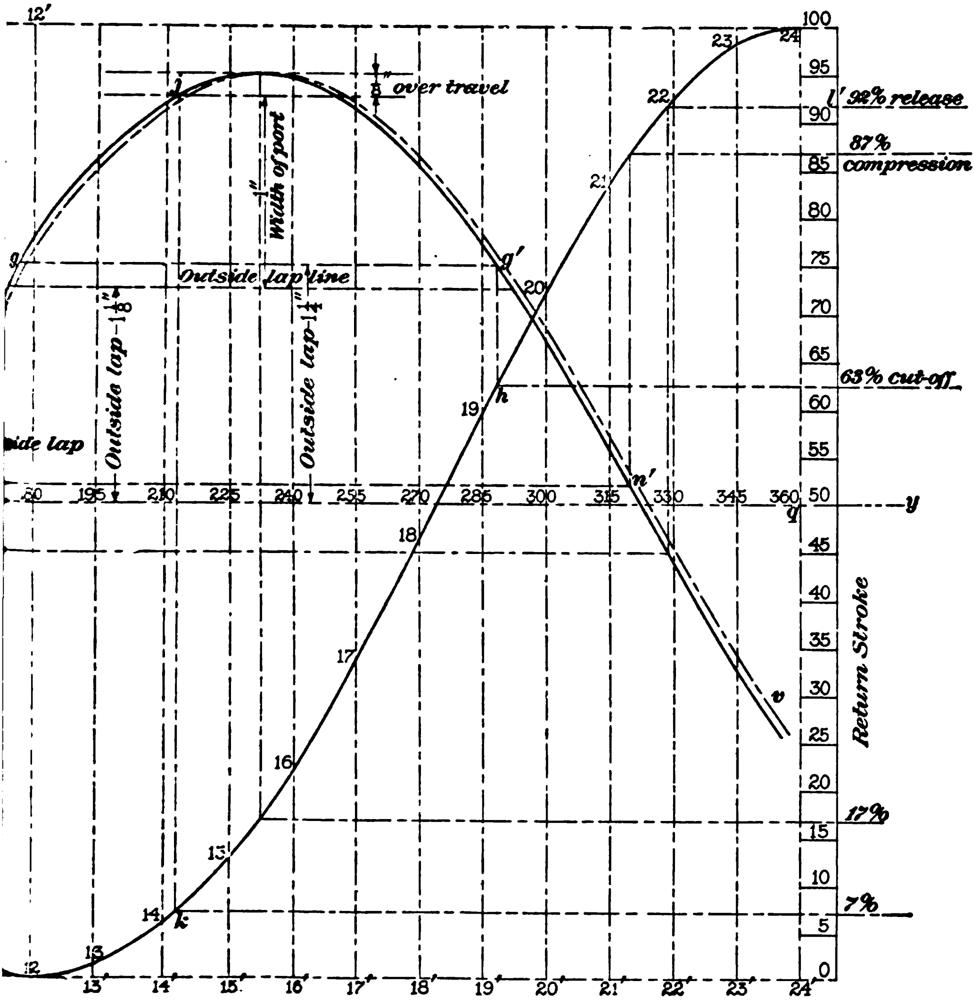
HARMONIC DIAGRAM CONSIDERING ANGULARITY OF CONNECTING-ROD

39. Piston Displacement Curve.—To lay out the harmonic diagram for an engine with a connecting-rod, take a base line *xy*, Fig. 24, and on it lay off *pq* of any convenient length, but preferably some even dimension. The length of this line *pq* represents the angle turned through by the crankpin in one revolution, and should be divided into equal parts representing a given number of degrees. A convenient division is 15°, or the whole length of the line is divided into $360 \div 15 = 24$ parts, as in Fig. 23. A height should now be chosen to represent the length of the stroke of the piston to some convenient scale and half of this height should be laid off on each side of *xy*, as indicated by *uv*. This line may then be conveniently divided into percentages of the stroke, as shown.

The next step is to find the displacement of the piston for each 15° of motion of the crank. This is most readily accomplished graphically, as shown in Fig. 25. Draw the crankpin circle *ab* to any convenient scale and divide it into the same number of equal parts as there are divisions on the line *xy* of Fig. 24; or simply draw the half crankpin circle and divide it into one-half the number of divisions into which *xy* was divided. Then, with a radius equal to the length of the connecting-rod, to the same scale, and with the points on the crankpin circle as centers, draw arcs cutting the line *ab* extended in the points *1*, *2*, *3*, etc., as shown. Mark the points on the circle and on *ab* extended with corresponding

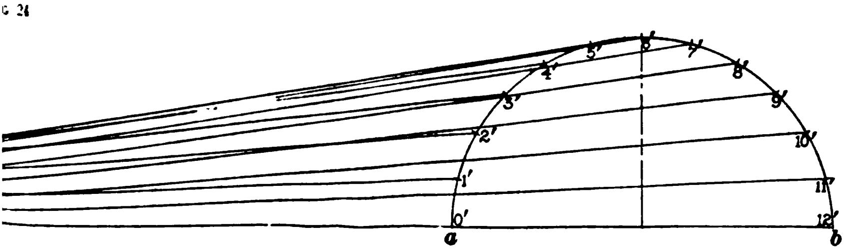

Head End
0 u 1′ 2′ 3′ 4′ 5′ 6′ 7′ 8′ 9′ 10′
Forward Stroke
22%
i
Angular advance
53°
Valve displacement curve
Piston displacement curve
Outside lap
Inside lap
69% cut-off
Outside lap line
Width of port
87% compression
92% release
Crank End


Head End
Return Stroke
Crank End
0 1 2 3 4 5 6 7 8 9 10 11 12
Forward Stroke



$\frac{1}{8}$" over travel
1" Width of port
Outside lap line
Outside lap-$1\frac{1}{8}$"
Outside lap-$1\frac{1}{4}$"
l' 92% release
87% compression
63% cut-off
17%
7%
Return Stroke

G. 24



G. 25

numbers. The points on *a b* extended will then indicate the relative piston positions for corresponding positions of the crankpin. These points are now transferred to the vertical lines through the corresponding degree divisions on the line *x y*, as shown on the diagram, Fig. 24. It will be found convenient, when possible, to take the length of the stroke of the engine to the same scale in both Figs. 24 and 25; otherwise it becomes necessary to change the scale in transferring dimensions from one to the other.

40. Starting with the line *u v*, Fig. 24, passing through the 0° position on *x y*, take the first position at *O*, since the piston is at the end of the stroke. On the 15° line, lay off *1′–1* equal to *0–1* in Fig. 25, making allowance, of course, for the different scales of the drawings, if the scales are not the same; on the 30° line, Fig. 24, lay off *2′–2* equal to *0–2* in Fig. 25, and in like manner lay off the other points, until the 180° position is reached, at which point the piston is at the opposite end of its stroke. On the return stroke, the positions of the piston are the same as on the forward stroke; that is, for such positions of the crankpin below the center line there is a corresponding position above the center line. For the 195° position, therefore, take *13′–13*, Fig. 24, equal to *12–11* in Fig. 25, and for the 210° position make *14′–14* equal to *12–10* in Fig. 25, etc. These points may be obtained more easily, however, by simply projecting the piston positions on the first twelve ordinates to the ordinates for the twelve corresponding portions of the return stroke. That is, point *1* may be projected over to point *23*, point *2* to *22*, point *3* to *21*, etc. A curve should now be drawn through the twenty-four points *1*, *2*, *3*, *4*, etc. thus found.

On the left-hand edge of Fig. 24 is now drawn, as shown, a scale of percentages of the stroke in 5-per-cent. divisions. A horizontal line from any point on the curve, as *1*, *2*, *3*, etc., to this scale, shows at once what percentage of the stroke the piston has covered from the starting point for that crank position. A similar scale is drawn on the right-hand side for the return stroke, starting at the bottom and reading up. The projections of any point on the curve *1*, *2*, *3*, etc. upon the scale of percentages on the left, and the scale of degrees on xy, will show the exact displacement of the piston and the corresponding position of the crank-pin. The curve *1*, *2*, *3*, etc. is called the **piston-displacement curve.**

41. Valve-Displacement Curve.—The next step is to draw a similar curve on a good quality of tracing paper, representing the displacements of the valve for any angle turned through by the eccentric. If the eccentric rod is very short, or if extreme accuracy is desired, this curve should be drawn by the same method as that used in finding the piston-displacement curve of Fig. 24; otherwise it may be drawn as in Fig. 23.

Having drawn the two curves, the next step is to place the one on tracing paper over the other, so that both can be seen. Then, when the valve-displacement curve is properly placed over the piston-displacement curve, they will at all times correctly represent the positions of piston, crank, valve, and eccentric center. To locate the valve curve properly, the line dd' is drawn on the tracing paper with the valve curve, and to the same scale, at a distance cd below xy equal to the outside lap. Below the line dd', a distance de equal to the lead is then laid off, and ee' is drawn parallel to dd', intersecting the valve curve at e'. It is evident that the distance of the point e' from xy is equal to the lap plus the lead, which is the displacement of the valve when the piston is at the beginning of its stroke, as explained in connection with Fig. 23. Hence, place the tracing on the piston-curve drawing, so that the axis xy of the one lies on the axis xy of the other, and so that the point e' of the valve curve falls on the line uv of the piston curve. The two curves are then in their relative positions to give the desired head-end lead.

42. Angle of Advance.—The angle of advance may also be measured directly from this diagram. Since the angle between the crank and eccentric radii is equal to 90° + the angle of advance, it is apparent that the valve reaches the end of its travel before the crank reaches the 90° position by an angle equal to the angle of advance. By measuring the angle between the point at which the valve is in its extreme position and the 90° position of the crank, the angle of advance will be found. Draw the line fi, Fig. 24, from the point f of maximum valve displacement, at right angles to xy. Then the angle represented by the distance from this line to the 90° line will be the angle of advance, which in this case is about 38°. The outside lap on the head end was taken at $1\frac{1}{4}$ inches. If the outside lap on the crank end is taken as $1\frac{1}{4}$ inches, and the line gg' is drawn $1\frac{1}{4}$ inches above the base line, then by projecting the point g', at which the valve closes, on the piston curve at h, and thence to the scale of percentages on the right, it will be found that cut-off takes place at about 63 per cent. of the return stroke. Similar projections for the forward stroke will show that the cut-off does not occur until 69 per cent. of the stroke.

To give more nearly equal cut-offs, the lap on the crank end might be reduced to $1\frac{1}{8}$ inches, which would make a later cut-off on the return stroke. This, however, would increase the crank-end lead to $\frac{1}{4}$ inch, which is excessive, and it would therefore be necessary to turn the eccentric backwards on the shaft, decreasing the angle of advance and also the lead at both ends. If the angle of advance were thus decreased until the leads were $\frac{1}{16}$ inch and $\frac{3}{16}$ inch at the head and crank ends, respectively, the cut-off would occur at 70 per cent. on the forward stroke and at 68 per cent. on the return stroke, which is close enough for all ordinary purposes. The dotted curve tuv shows the position of the valve-displacement curve to give the conditions just mentioned.

When it is desired to study the effect of varying the angular advance, the valve curve on the tracing paper should be moved to the left to increase the angular advance and to the right to decrease the angular advance, and when the final positions are settled upon, the valve curve should be pricked through and thus transferred to the piston-curve sheet for permanent record.

43. Overtravel of the Valve.—Decreasing the lap also gives opportunity to study another important point that is clearly brought out by the harmonic diagram, namely, the effect of overtravel of the valve. Consider the forward stroke first; the valve reaches its maximum displacement at the point *f*, which also coincides with full port opening; now, projecting the point *f* up to the piston curve at the point *i*, and thence to the scale at the left of the diagram, it is found that the port is wide open at about 22 per cent. of the stroke of the piston. Now consider the return stroke; assuming a maximum valve displacement of $2\frac{1}{4}$ inches and a width of port of 1 inch, and laying off the port opening from the $1\frac{1}{8}$-inch lap line, as shown, it is found that the valve when in its outer position has traveled $\frac{1}{8}$ inch beyond the outside edge of the port, thus giving an overtravel of $\frac{1}{8}$ inch. Since the valve has $\frac{1}{8}$-inch overtravel, due to decreasing the outside lap, the full port opening occurs when the valve-curve line cuts the back edge of the port at the point *j*. Projecting the point *j* down to the piston curve at *k*, and thence to the right-hand percentage scale, it is found that a full port opening occurs on the return stroke when the piston has traveled only about 7 per cent. of its stroke. If the point of maximum valve displacement is projected upon the piston curve and then upon the right percentage scale, it is found that with no overtravel the full port opening would not be reached until the piston had traveled about 17 per cent. of its stroke. It will be seen, therefore, that a little overtravel is very valuable in giving a full, clear port opening early in the stroke.

44. Port Opening.—The distance that the port is open at any instant can be found by simply projecting the desired point on the percentage of stroke scale to the piston curve, then down or up, as the case may be, to the valve curve, and measuring the intercept between the valve curve and the lap line. For example, if it should be desired to find the width of the port opening when the piston is at 22 per cent. of its forward stroke, project the 22-per-cent. point to the point *i* upon the piston curve, and then project the point *i* down upon the valve curve at *f*. The vertical distance between the point *f* and the outside lap line *d d'* represents the width of the port opening at that instant.

45. Release and Compression.—Assume that a compression of about 15 per cent. of the stroke of the piston is required. For the case in hand, the point of release is taken as $\frac{11}{12}$ stroke. To determine the amount of inside lap to give this release and compression, it is necessary simply to locate the $\frac{11}{12}$ point on the forward stroke, as *l*, and project this point horizontally to the piston curve at *m*, then up to the valve curve at *n*. The distance from the base line *x y* to the point *n* then represents the inside lap. Projecting the point of release *n* horizontally to the point *n'* on the opposite side of the valve curve, at which point the exhaust closes, then up to the piston curve and out to the right-hand percentage scale, as shown, it is found that the exhaust is closed at 87 per cent. of the stroke, thus giving a compression of 13 per cent. The amount of the inside lap measures $\frac{3}{32}$ inch. The same amount of inside lap on the return stroke may, by a similar process, be found to give compression at about 90 per cent. and release at 90 per cent. In order to determine the inside lap required for the return stroke to cause release to take place at $\frac{11}{12}$ stroke with a 13-per-cent. compression, lay off on the right-hand percentage scale $\frac{11}{12}$ stroke, as shown at *l'*. Project this point horizontally to the piston curve, and then vertically to the valve curve. The point at which it meets the valve curve lies below the base line *x y* a distance of $\frac{1}{4}$ inch, which represents the inside lap. Projecting this point over to the other side of the valve curve, then down to the piston curve and over to the percentage scale, it will meet the latter at 87 per cent. of the stroke and give a compression of 13 per cent., as required. It will be seen, therefore, that $\frac{1}{4}$-inch inside lap is required on the return stroke to give the same points of release and compression that are given by $\frac{3}{32}$ inch on the forward stroke.

VALVE GEARS

(PART 2)

VARIABLE CUT-OFF VALVES

EXPANSION VALVES

1. Meyer Cut-Off Valves.—In designing slide valves, the engineer may meet with conditions that are difficult, if not quite impossible, to overcome without using independent expansion or cut-off valves. With the advent of high-pressure steam came the recognition of the economy resulting from high ratios of expansion and the needful early point of cut-off, for which the plain slide valve could not be used. For, with the plain slide valve, it is not feasible to obtain a cut-off earlier than $\frac{1}{2}$ stroke, the usual point of cut-off with this type of valve being at about $\frac{3}{4}$ stroke. To obtain a cut-off between $\frac{1}{2}$ and $\frac{3}{8}$ stroke, the laps must be increased, necessitating longer valve travel, higher compression, and earlier release, requiring greater force to move the valve, and resulting in added wear. But, by the use of a separate and independent cut-off valve, the point of cut-off can be adjusted to give any desired ratio of expansion. The main valve can then be designed to suit the requirements of admission, release, and compression.

The most common and best known type of expansion or cut-off slide valve is the **Meyer valve**, which consists of a pair of adjustable plates or cut-off valves, bearing on the back of a main valve. The main valve is provided with through ports that are opened and closed by the plate of the expansion valve. This type of valve is frequently used on the steam cylinders of air compressors and pumps.

Fig. 1 shows a good arrangement for a Meyer valve. The main valve *a*, in this case, is driven by the main eccentric in the usual way, the point of cut-off generally chosen for the main valve being at $\frac{3}{4}$ stroke. The cut-off valves *b*, *b* are

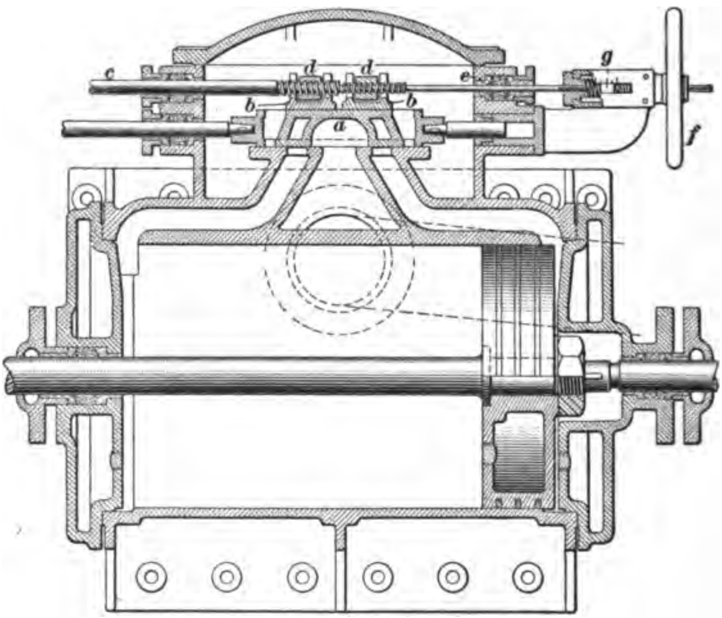


FIG. 1

driven by a separate eccentric, generally placed 180° ahead of the crank, though not necessarily so.

By means of right-hand and left-hand screws on the cut-off valve stem *c* and the nuts *d*, the cut-off valves are either spread apart or drawn together, as an earlier or later cut-off is desired. The gearing is generally so arranged that this adjustment of the cut-off valves can be accomplished while the engine is running. On the cut-off valve stem *c*, which extends through the outer steam-chest wall, as shown at *e*, is placed a hand wheel *f*, by means of which the stem may be rotated. A suitable indicator or scale *g*, mounted on the hand-wheel bracket serves to show at what point of the stroke cut-off is taking place. The motion of the Meyer valve can best be studied by means of the harmonic valve diagram.

2. The **harmonic valve diagram** provides an excellent means of analyzing the action of the Meyer cut-off valve. It is desired to obtain an early cut-off with this gear, and in order to observe its action as compared with that of the plain slide valve, the piston displacement curve and the main valve curve are drawn, as in Fig. 2, which represents a harmonic diagram for the study of the action of the Meyer cut-off valve. It will be seen that the piston displacement curve, the main-valve curve, and the cut-off valve curve are all drawn on one sheet, the center lines *x y* of the three coinciding so as to form one line. The main-valve eccentric is given 37° angular advance, while the cut-off valve has its eccentric 180° ahead of the crank.

This position corresponds to an angular advance of 90°. The two eccentrics are assumed to have the same throw. Inasmuch as the main valve does not regulate the cut-off, its outside laps are made equal, each being $1\frac{1}{4}$ inches. On the diagram, it will be observed that the outside lap line crosses the main-valve curve at *a*. Projecting this point vertically to *b* on the piston displacement curve, and thence horizontally to the left-hand percentage scale, it is found that cut-off by the main valve would occur at 68 per cent. of the forward stroke. Similarly, projecting the point of cut-off, *c*, on the crank end, to *d*, on the piston curve, and thence horizontally to the right-hand percentage scale, it is found that cut-off by the main valve would occur at 61 per cent. of the return stroke. These values, then, represent the latest possible points of cut-off on the two strokes, with the valve dimensions given.

Assume the compression at each end to occur at 87 per cent. of the stroke, corresponding to a compression of 13 per cent. The 87-per-cent. points are located on both percentage scales. These points are then projected horizontally to the piston

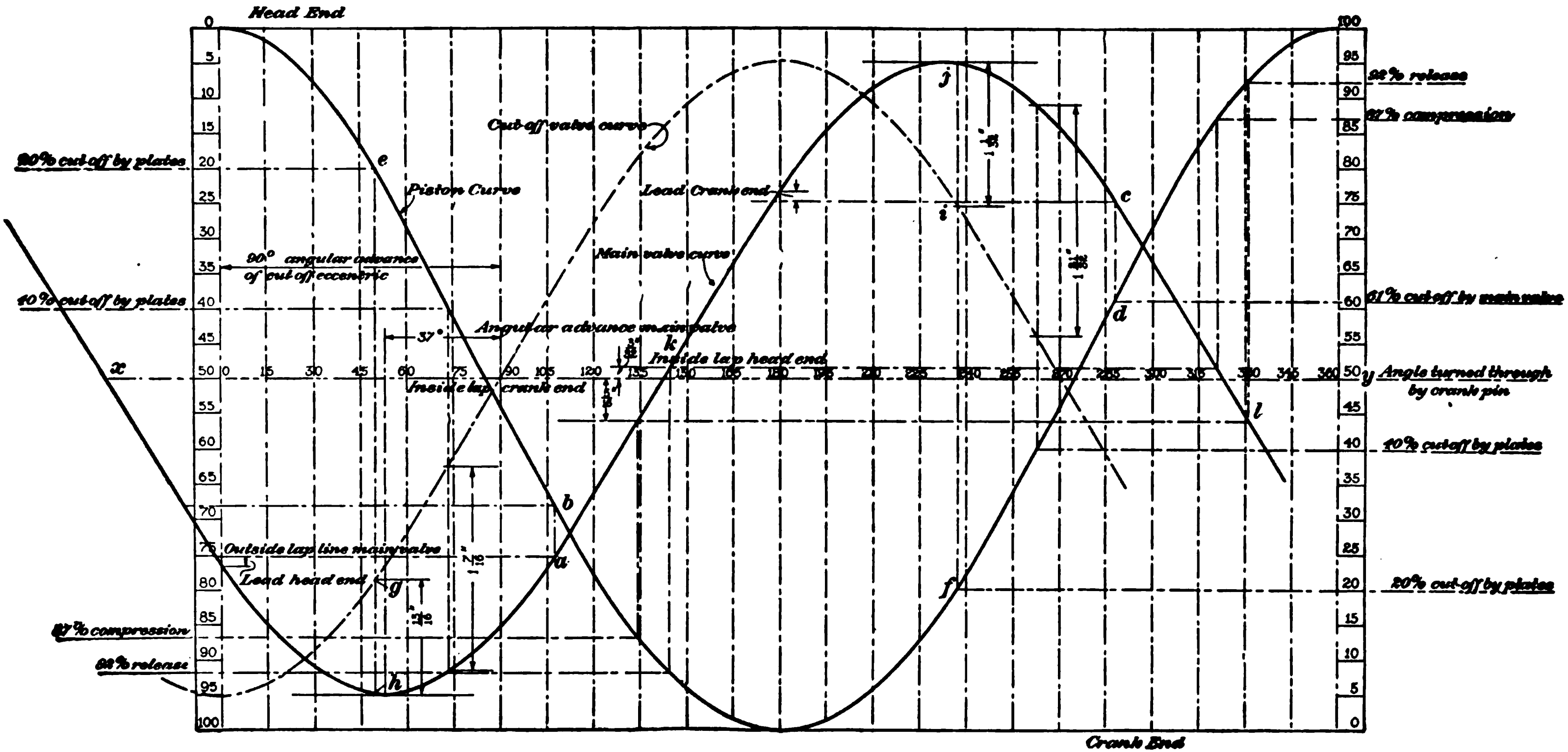

Head End
Crank End
Piston Curve
Cut-off valve curve
Main valve curve
Lead Crank end
Lead head end
Outside lap line main valve
Inside lap head end
Inside lap crank end
Angular advance main valve
90° angular advance of cut-off eccentric
37°
20% cut-off by plates
40% cut-off by plates
10% cut-off by plates
Angle turned through by crank pin

Fig. 2

curve, and thence vertically to the main-valve curve, showing that inside laps of $\frac{3}{32}$ inch on the head end and $\frac{5}{16}$ inch on the crank end are necessary. If these inside lap lines are extended, as shown in Fig. 2, so as to intersect the main-valve curve at *k* and *l*, and these points are then projected vertically to the piston curve, and thence to the percentage scales, it will be found that release occurs at 92 per cent. of each stroke. The outside laps being the same for both ends, the leads also are equal.

3. Let it be assumed that cut-off is to take place at 20 per cent. on each stroke by the use of the cut-off valves. First, the 20-per-cent. points are located on the percentage scales and are projected horizontally to the piston displacement curve at *e* and *f*. From *e*, a vertical line is drawn,

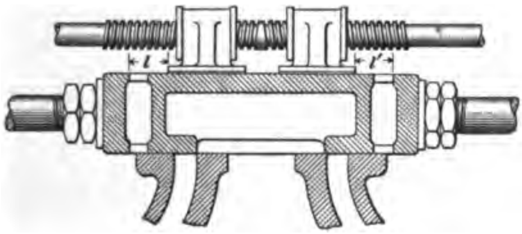


FIG. 3

cutting the two valve curves at *g* and *h*, while a similar vertical line from *f* cuts the two valve curves at *i* and *j*. It is evident, then, that the distance *g h* represents the difference of the valve displacements at cut-off on the forward stroke, and that *i j* represents the difference for the return stroke. In other words, if the center lines of the two valves were placed so as to coincide, the edges of the cut-off plates would be at the distances *g h* and *i j* from the outer edges of the openings in the main valve.

Hence, to set the two valves in the actual engine so as to give correct cut-off according to the layout in Fig. 2, they are so placed that their center lines coincide, as in Fig. 3. The head-end cut-off plate is then shifted until the distance *l* equals *g h*, Fig. 2, and at the other end *l'* equals *i j*, Fig. 2. It is assumed, of course, that the main valve has been

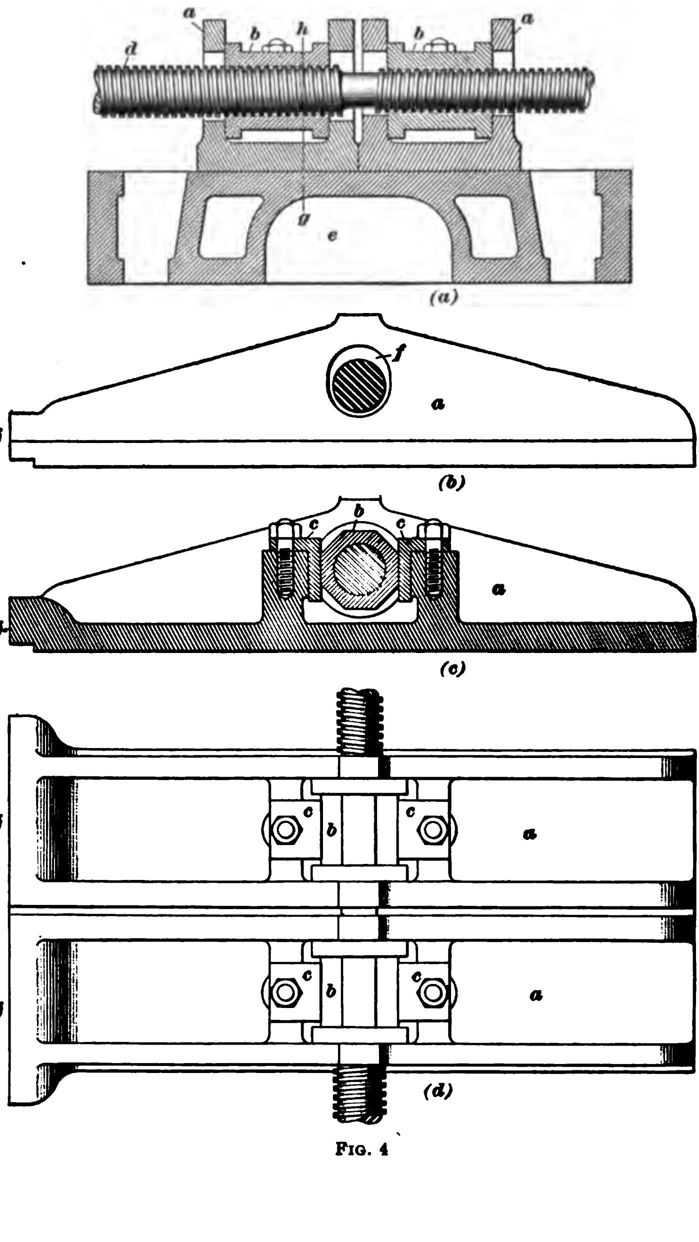


FIG. 4

designed and set correctly, and its eccentric fixed, according to the data given in Art. 2.

4. If it is desired to have the cut-off occur at $\frac{4}{10}$ stroke, the 40-per-cent. points are located and projected to the valve curves, as in the previous paragraph. Instead of obtaining intercepts of $\frac{13}{16}$ inch and $1\frac{1}{32}$ inches, which are the values of $g\,h$ and $i\,j$, the intercepts for 40-per-cent. cut-off are found to be $1\frac{7}{16}$ inches and $1\frac{21}{32}$ inches, respectively. However, the difference between $1\frac{7}{16}$ and $1\frac{21}{32}$ is exactly the same as the difference between $\frac{13}{16}$ and $1\frac{1}{32}$, namely, $\frac{7}{32}$ inch; and for all points of cut-off between 20 per cent. and 40 per cent. the difference of the intercepts at the head and crank ends is practically constant. Therefore, the cut-off may be varied between 20 per cent. and 40 per cent. by simply screwing the cut-off valve plates closer together, without resetting them, since they have already been set to give a difference of $\frac{7}{32}$ inch between l and l' at 20-per-cent. cut-off. At 40-per-cent. cut-off, the values of l and l' will be $1\frac{7}{16}$ and $1\frac{21}{32}$ inches, respectively. For points of cut-off earlier than 20 per cent. or later than 40 per cent., the difference between l and l' will not remain constant, but will vary slightly.

5. **Details of the Meyer Valve.**—To provide for the adjustment of the Meyer cut-off valves in accordance with the difference in the expansion required, the cut-off valve plates *a, a*, Fig. 4 (*a*), are usually designed with nuts *b, b*, so arranged that by taking out loose gibs *c, c*, Fig. 4 (*c*), at the sides, the nuts can be rotated and each valve adjusted independently of the other. Fig. 4 (*a*) shows a section of the main valve *e* and the expansion valves *a, a* on the center line of the valve stem *d*; it also shows the sizes of the stem and the right-hand and left-hand threads by means of which the cut-off is adjusted. Fig. 4 (*b*) shows an elliptic opening *f* through the cut-off valve for the stem, thus allowing the valve to seat itself freely. Fig. 4 (*c*) shows a cross-section taken along the line *g h*, Fig. 4 (*a*), and Fig. 4 (*d*) shows a plan of the cut-off valves. When the valve seat is horizontal, both ends of the valves are made as shown at *i*;

but when the seat is vertical, the lower edges are provided with bearing surfaces, as shown at *j*, that slide on corresponding surfaces in the valve chest, thus relieving the valve stem of the weight of the valve. The Meyer valve gear gives an excellent distribution of steam through the most economical ranges of cut-off for slide valves. The force required to move the main valve, however, is considerably greater than that required for the common slide valve, since it is necessary to overcome the friction between the main and the cut-off valves in addition to the friction between the main valve and the valve seat.

6. Balancing the Meyer Valve.—Many attempts have been made to balance the Meyer valve, one of which is

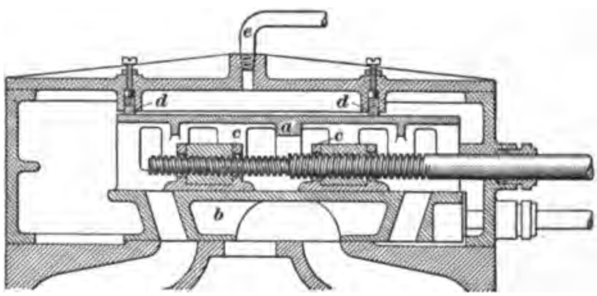


FIG. 5

illustrated in Fig. 5. This arrangement consists of a frame *a*, which forms a part of the main valve *b* and extends up over the top of the cut-off valves *c*, *c*. A cast-iron ring *d*, which is fitted into a groove turned in the steam-chest cover, bears on the back of this frame. The ring is held out against the back of the valve by short helical springs. The chamber within the balance ring is connected to the exhaust by the pipe *e*. The main valve is thus relieved of a load equal to the difference between the steam and exhaust pressures per square inch multiplied by the horizontal area within the balance ring.

7. The Rider Valve.—Another type of valve, resembling the Meyer valve in its action, and known as the **Rider valve,** is illustrated in Fig. 6 (*a*). It is cylindrical in form, as shown at *a*, and has its seat in the back of the main valve *b*, in which the ports are arranged in the form of helixes, as shown at *c*, *c*, Fig. 6 (*b*). The edges of the cut-off valve form similar helixes, as shown at *e*, *e*, Fig. 6 (*d*), in which is shown also a section of the valve *b*. The development of the valve seat is shown at *f*, Fig. 6 (*c*). It is

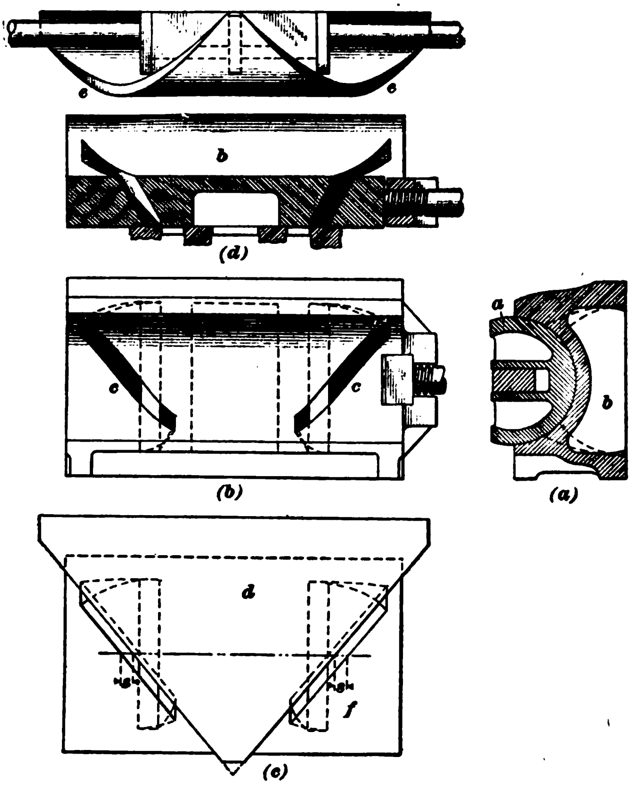


FIG. 6

evident that rotating the valve is equivalent to moving a wedge-shaped valve, as *d*, up or down, and thus increasing or decreasing the distance *s* from the cut-off edge of the cut-off valve to the cut-off edge of the main valve, measured along their line of motion, when the valves are in their central positions.

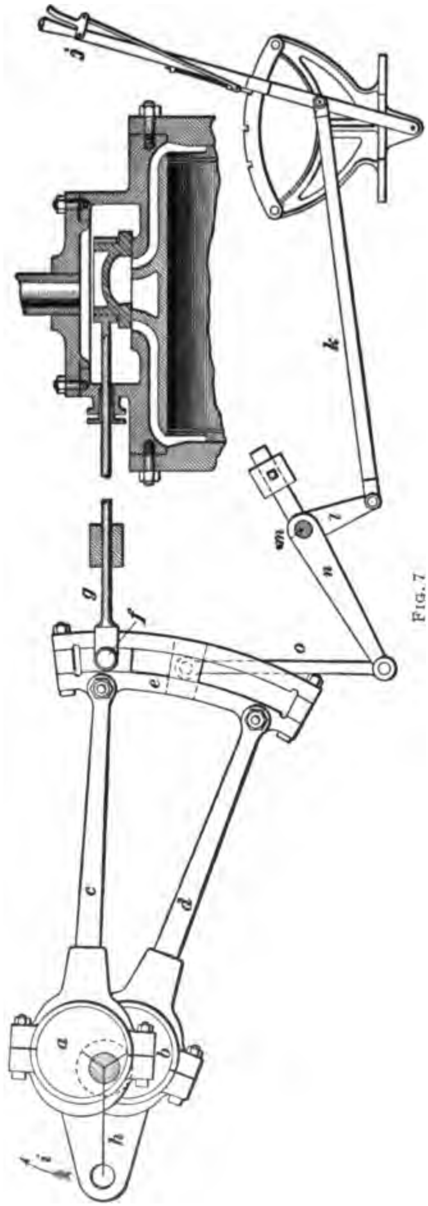


FIG. 7

This is similar to separating or drawing toward each other the cut-off valves or plates of the Meyer valve, and thus producing an earlier or a later cut-off. The action of this valve can also be investigated by means of the harmonic diagram; in fact, the same diagram will answer for both the Rider and the Meyer valves when the conditions are the same for both. The Rider valve has the advantage over the Meyer valve that the force required to turn the valve is very small, which, therefore, can be easily adjusted by a governor.

REVERSING GEARS

LINK MOTION

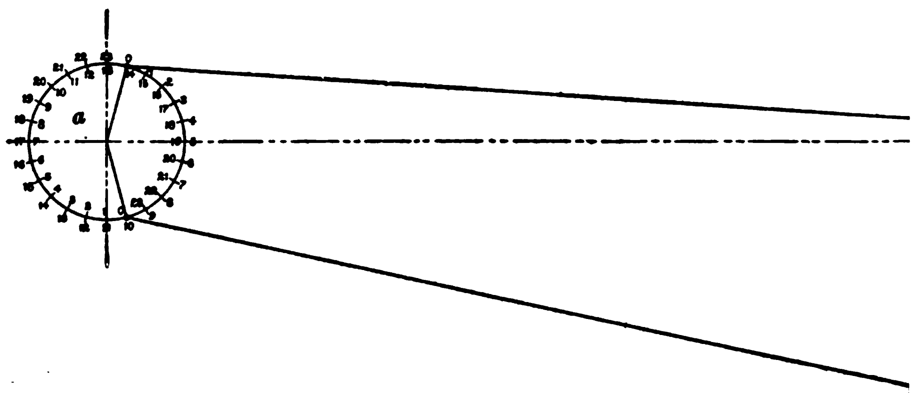
8. The Stephenson Link Motion.—The most common reversing gear, is the **Stephenson link motion,** shown in Fig. 7. Two eccentrics, *a* and *b*, are used, one for the forward motion and one for the backward motion of the engine. The extreme ends of the eccentric rods *c*, *d* are connected to a curved bar or link *e*, which engages a block or pin *f* on the valve stem *g*. Since the valve is driven directly by the eccentric, without a reversing rocker, the crank *h* must follow the eccentric for either direction of motion. An engine is generally said to run forwards, or over, when the crankpin sweeps through the upper half of its circle while the piston is moving toward the crank or when the piston rod moves out of the cylinder. Then the eccentric *b* must operate the valve when the engine runs forwards, or over, and eccentric *a* when it runs backwards, or under, which is the direction shown by the arrow *i*. When the link is in the lower position and the rod *c* lies nearly in a straight line with the valve stem *g*, the valve receives the motion imparted by the backward eccentric *a*; and when the link is in its extreme upward position, the valve receives the motion imparted by the forward eccentric *b*.

9. The link motion, it will be seen, thus gives a ready means of reversing the engine. By putting the link in an intermediate position, the valve is given a motion nearly the same as would be produced by an eccentric of less throw and increased angular advance. The effect of this change is to hasten the cut-off and the compression, and to increase the ratio of expansion. Hence, it will be seen that the link may be used either to vary the cut-off or to reverse the engine. When the link is in mid-position, that is, when the block *f*, or valve-stem connection, stands midway between the points where the eccentric rods are connected to the link, very little work is done, as the valve then has a total displacement equal to the lap and lead only.

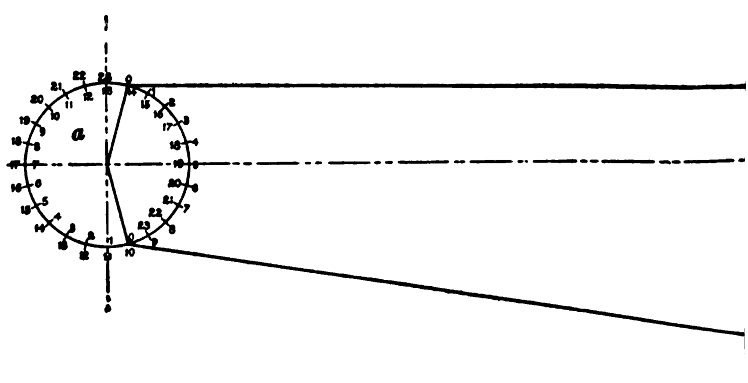
In small engines, the vertical movement of the link is effected by a hand lever *j* connected by a rod *k* and arm *l* to a tumbling shaft *m*, which is joined to the reversing link *e* by means of an arm *n* and the link, or lifting rod, *o*. The link *o* stands at one side of the reversing link, thus allowing the latter to swing in the plane of motion of the valve. As reversing engines are built in pairs, set side by side, there are two links to be thrown or shifted simultaneously by means of the shaft *m*.

10. Laying Out the Reversing Gear.—The layout of the valve and eccentric for a reversing gear is similar to that for the plain slide valve, and the methods given in *Valve Gears*, Part 1, may be used. There is, however, this difference, that the point of cut-off when in full gear should be chosen at about 85 per cent. of the stroke, and the over-travel of the valve should be 25 per cent. of the width of the port; that is, if the steam port is 1 inch wide, the valve should travel over the inner edge about $\frac{1}{4}$ inch. Having settled on the type and length of the link, the point of cut-off, etc., when the valve is in full gear, it becomes necessary to determine the link radius. This may be equal to the distance from the center of the shaft to the center of the link-block pin when in mid-position on the link, although it is frequently made equal to the distance from the eccentric center to the center of the link block when in full gear, that is, the distance from the center of *a* to the center of *f*, Fig. 7. The lifting rods should be made as long as the conditions will allow. The next consideration is the point of suspension for the upper end of the lifting rods, for the slip of the link depends on the location of this point. The slip is the vertical motion of the link on the block due to its swinging about the point of suspension of the lifting rod.

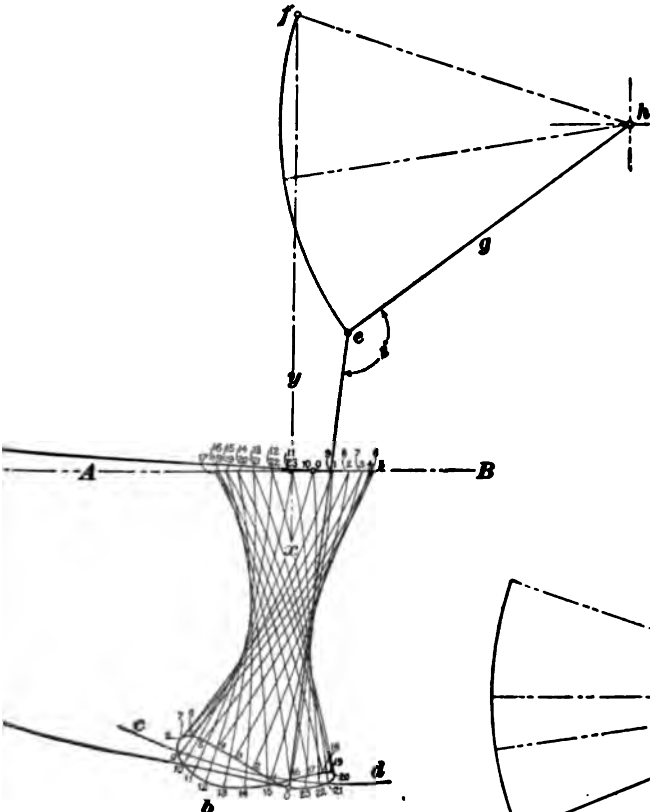
To determine the slip, lay out a skeleton diagram of the connections, as shown in Fig. 8, dividing the eccentric circle *a* into any number of equal parts. Then, preferably by means of a templet, draw the center line of the link in the several positions corresponding to the various divisions of the eccentric circle, assuming that the end of the link in which the end of the block will generally run moves in a straight line. The opposite end of the link will describe a curve *b*, resembling a distorted figure **8**. Next, with a radius equal to the length of the lifting rod, describe the arc *c d*, with the center *e* so located that the arc *c d* will divide this figure **8** in

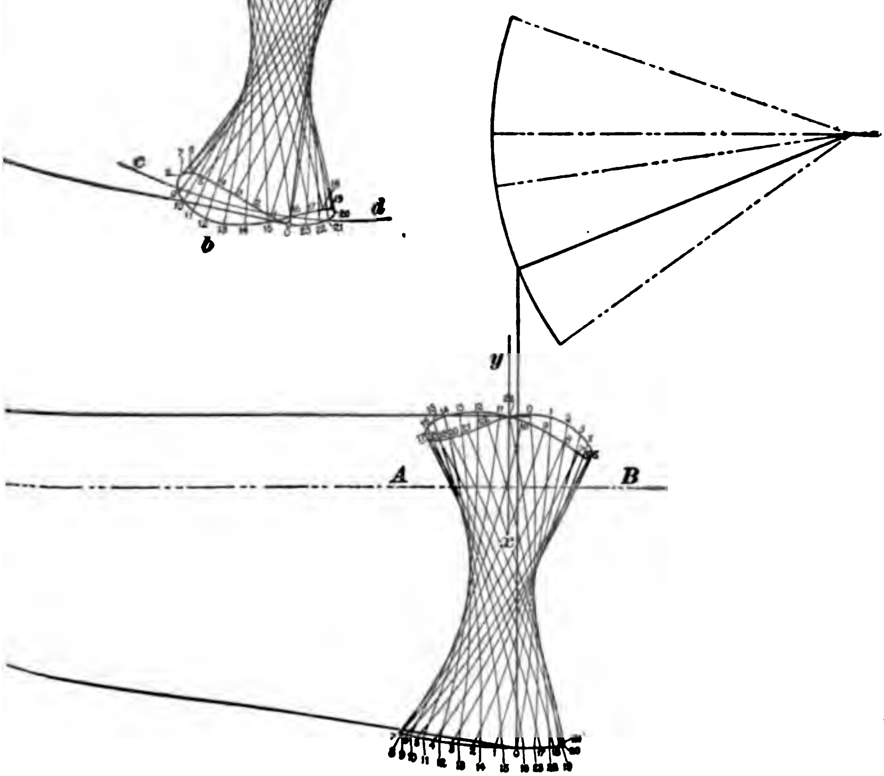


Fi



Fi


f
h
g
e
y
A
B
x
c
d
b


y
A
B
x

such a manner that the extremes of the curve will be equally distant from the arc on each side. The greatest distance across the figure **8**, measured along the lifting rod, will then represent approximately the slip of the block in the link, and the greatest distance from the arc *c d* to a point on the figure **8** is the least clearance necessary at the ends of the link.

The center *e* thus located is the best position for the point of suspension of the hanger rod. The point *f* for the upper center of suspension should be vertically over the upper mid-position of the link-block pin and at such a distance up that the link-block pin will vibrate equally on each side of the vertical position of the lifting rod. The length of the reversing arm *g* is determined by the angle through which the tumbling shaft *h* turns. Care should be taken that the angle *i* does not become too large when the link is in its extreme positions. When the engine is used for hoisting or for mill work, or in any service requiring the engine to run under and over alternately, the link should be suspended at its center, and the upper end *f* of the hanger should be located vertically over the mid-throw position of the link block.

Having determined the proportions of the valve, as lap, lead, travel, and overtravel, also the center lines of the link, its form and points of suspension, the only remaining feature to be investigated is the exact movements of the valve when the link is in various positions. This may be done by laying out a skeleton diagram of the link in three positions—full forward gear, quarter gear, and mid-gear. Fig. 9 shows this layout for quarter gear, that is, when the link stands one-quarter of its length from full gear. Then, by means of the harmonic diagram, the exact motion of the valve relative to the piston may be shown. It must be borne in mind, however, that the displacements of the valve for the twenty-four positions shown in Fig. 9 must be used in plotting the valve curve of the harmonic diagram. This may be done for any position of the link block, but if the displacement curve be laid out for the three positions—full forward, quarter, and mid-gear—the others may be determined approximately by inspection.

In constructing the valve curve of the harmonic diagram, the valve displacements must be measured from mid-position, as in all previous cases. Thus, in Figs. 8 and 9, the line xy represents the center line of the valve when in mid-position, and AB represents the line of motion of the valve. Hence, in Fig. 8, the displacement of the valve, when the upper eccentric is at *2* on a, is the distance of the point *2* from xy, measured along AB. The displacement for position *8* is the same, since *2* and *8* fall at the same point on AB. In Fig. 9, the link motion is at quarter gear. However, the displacements are measured along AB, as before. The displacement of the valve for any eccentric position is the distance from the line xy to the point on AB where the corresponding center line of the link cuts AB.

In large reversing engines, or in reversing engines using high steam pressure, some mechanical means must be provided to throw the links. In large marine engines, this is done by a worm-wheel and worm, actuated by a pair of small reversing engines, the operation of which is controlled from the reversing hand lever. In hoisting engines, it is accomplished by a steam cylinder whose piston rod is connected by means of links to the tumbling shaft; while in rolling-mill work, the steam cylinder is replaced by a hydraulic cylinder.

11. Equivalent Eccentric.—A much less accurate but more convenient solution of link-motion problems is reached by what is known as the **equivalent eccentric,** which is an imaginary single eccentric producing nearly the same displacements as the combined action of the two eccentrics for any given position of the link. The following method of finding the equivalent eccentric is frequently used. From the center of the shaft lay out the crank radius a, Fig. 10, the two eccentric centers b and c, and the link de for the link position desired. Then draw an arc of the eccentric circle through b and c. The points d and e are the centers of the pins by which the eccentric rods are connected to the link. Take g as the center of the link block in any position, and on the arc bc locate the point f, so that $bf : bc = dg : de$. Then hf represents the equivalent eccentric, both in radius and in angular position.

The position of the equivalent eccentric having been found, the motion of the valve may be analyzed, as in the case of a simple slide valve. As has already been stated, the equivalent-eccentric method of analyzing the valve motion is not accurate, and should be used only when a rough approximation is desired.

12. Occasionally, the eccentric rods are crossed, that is, the top eccentric is connected to the bottom of the link, and the bottom eccentric to the top of the link, when both eccentrics stand toward the link. This, however, is seldom

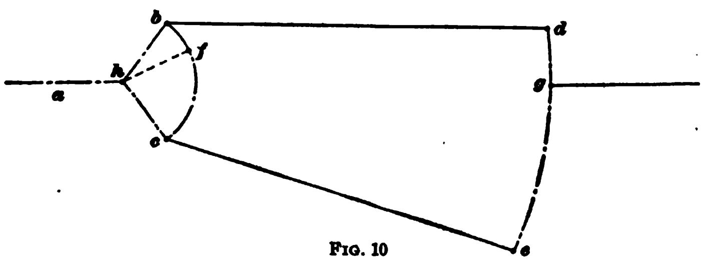


FIG. 10

done, and there is no advantage whatever in it, while the increased angularity of the rods causes undesirable variations in the motion of the valve. With crossed rods, the valve has no lead when the link is in mid-gear, and therefore it is sometimes said that crossed rods have the advantage of being able to stop the engine when the link is put in mid-gear. In practice, however, open rods also will stop the engine when the link is in mid-gear, as the amount of steam admitted is so small that it will not overcome the friction and compression. The action of valves with crossed rods may be investigated by means of the diagrams and methods already explained, following the same procedure as with the open rods, the only difference being that the eccentric rods are joined to the opposite ends of the link.

RADIAL VALVE GEARS

13. The Marshall Valve Gear.—Many devices have been invented for the purpose of reversing engines with one eccentric or without an eccentric, and to secure a better steam distribution than is possible with a link motion. Two

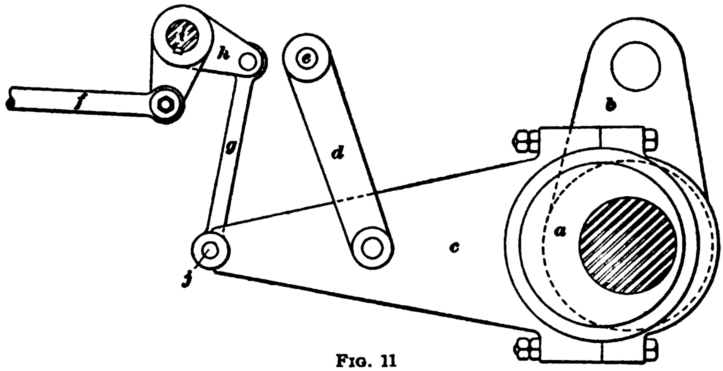


FIG. 11

of the best known of these and the two most used, are the *Marshall gear* and the *Joy gear*, both **radial valve gears.** Fig. 11 shows the general arrangement of a **Marshall gear** designed for a reversing rolling-mill engine. In this case, the eccentric *a* is placed at 90° to the crank *b*. The eccen-

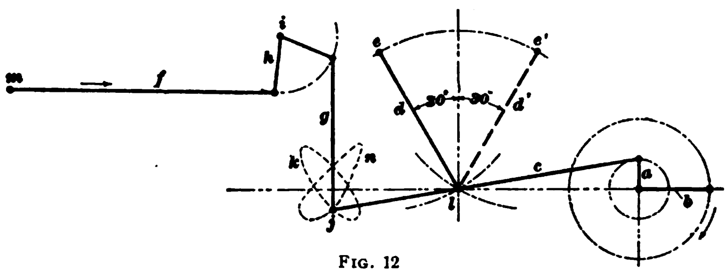


FIG. 12

tric rod *c* is constrained in its motion by the link *d*, pivoted at the point *e*, and is connected to the valve stem *f* through the link *g* and bell-crank *h*, which is keyed to the shaft *i* and swings about it. Fig. 12 shows a skeleton diagram giving the motions and various proportions of parts of the same gear, in which the reference letters indicate the same parts

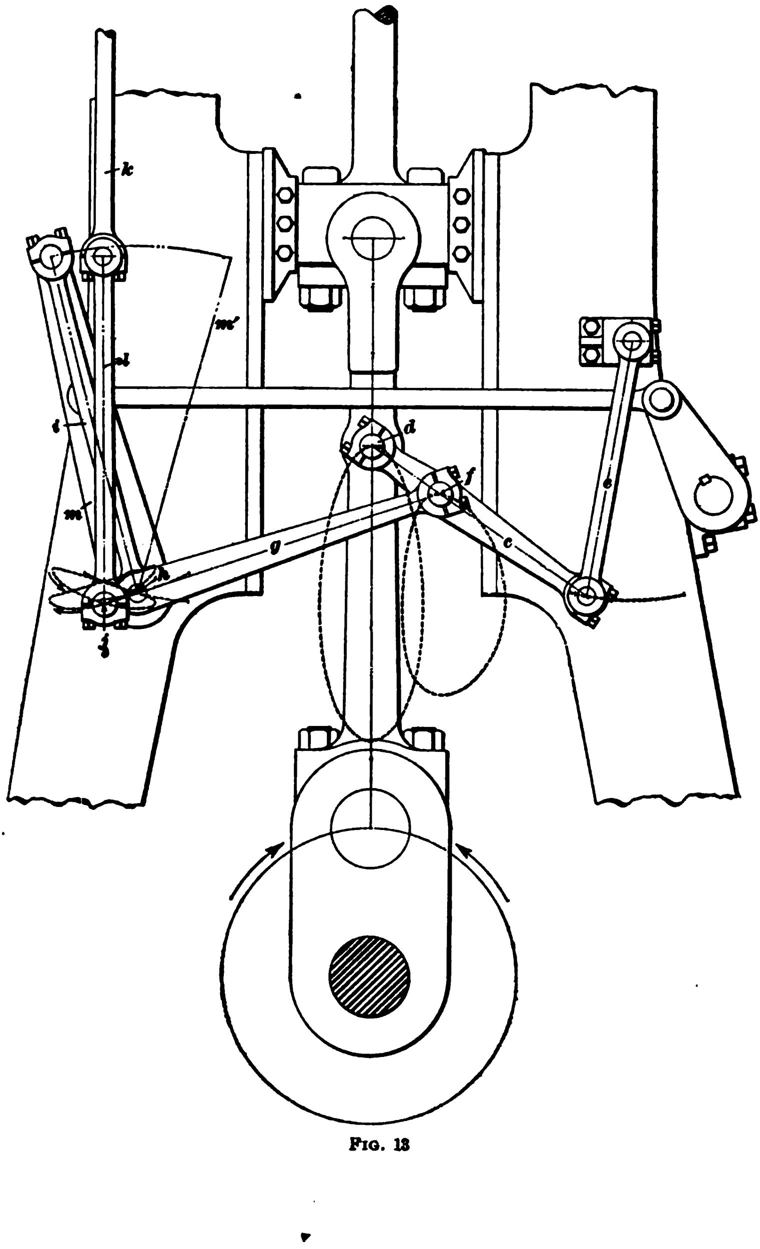


FIG. 13

as in Fig. 11. The link *d* is pivoted at *e* for one direction of motion, and for the opposite direction the center *e* is thrown over to *e'*. The dotted ovals *n* and *k* show the paths of the point of attachment *j* for both directions of motion. It will be noticed that the motion is taken from the eccentric rod *c* at a point *j* beyond the fulcrum *l*.

No definite rules can be given for the layout of Marshall gears, as they are generally used in cramped quarters and are limited by surrounding circumstances. The foregoing, however, indicates the method of determining the displacements of the valve that may be taken from the point *m*, and by means of the harmonic diagram the action of the valve can be fully investigated.

14. The Joy Valve Gear.—In the Joy gear, shown in Fig. 13, no eccentric is used, the motion of the valve being taken from a point on the connecting-rod and transmitted to the valve by means of a series of levers and links. A lever *c* is pivoted at one end to the connecting-rod at *d*, and the other end is attached to the link *e*. The upper end of the link *e* is pivoted on the frame of the engine, as shown. From the point *f*, on the lever *c*, a second lever *g* is carried to the side of the engine on which the valve mechanism is located, the fulcrum *h* being connected to the lower end of a link *i* or to a slide attached to the frame. The end *j* of the lever *g* is connected to the valve stem *k* by means of the link *l*. The upper end of the link *i* is connected to an arm *m*, carried at its lower end on a pin fixed to the engine frame, and free to be moved at the upper end by suitable linkages. This arm may, therefore, be thrown over to the position *m'*, or to any intermediate position. Throwing the arm from one position to the other reverses the engine. Or, if the center *h* slides in a guide, the angle of the guide may be changed to reverse the motion.

The proportioning of the parts of the Joy gear is best explained by means of the skeleton diagram shown in Fig. 14. This diagram illustrates the layout as made by the inventor. In this illustration, *a* represents the center of

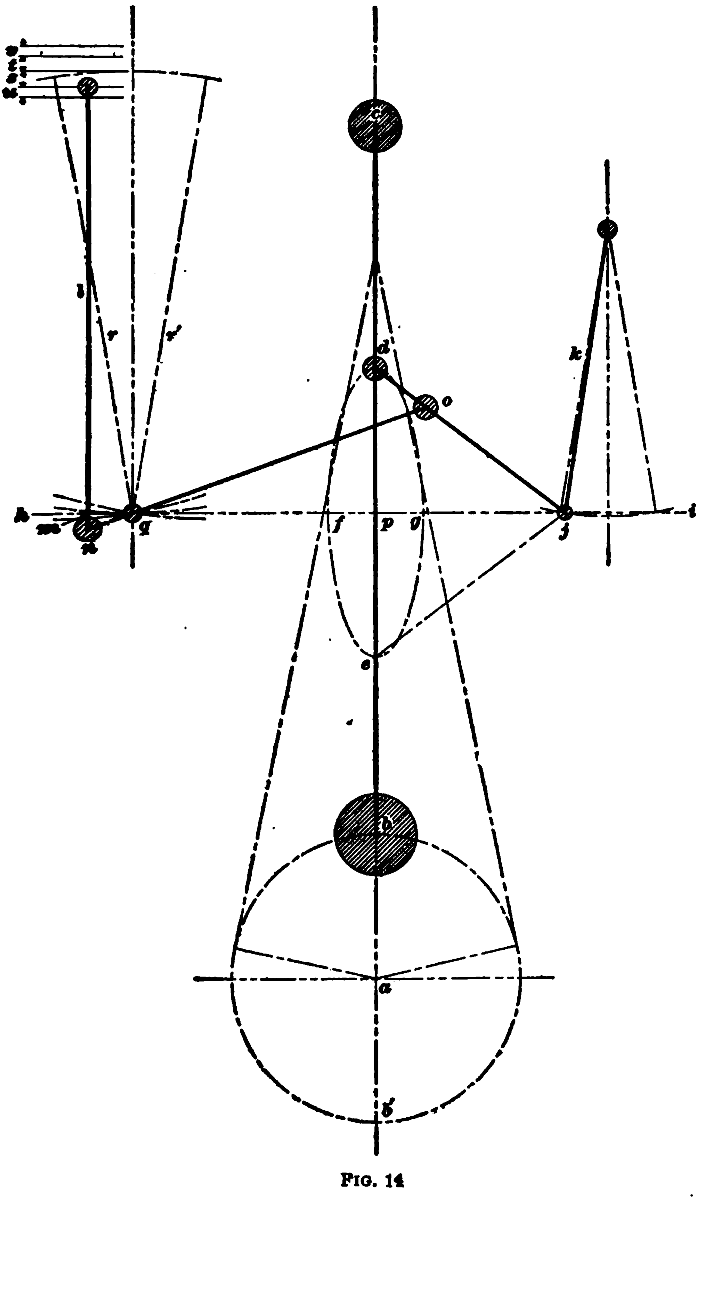


FIG. 14

the crank-shaft, $b b'$ the crankpin circle, and $b c$ the connecting-rod. A point d is taken on the center line of the connecting-rod at such a distance from the crosshead pin that it will describe an elliptic figure $d g e f$, the minor axis of which shall be at least twice the full travel of the valve. Through the extreme positions f and g draw the line $h i$, and from the extreme points d and e lay off the two lines $d j$ and $e j$, with the point j far enough from the center line of the engine to make an angle $d j e$ slightly less than a right angle. The point j is called the anchor point and should be guided as nearly in a straight line as possible, either in a guide or by a long anchor link k. On the center line of the valve rod l lay off the distance $m n$ vertically below $h i$, equal to the lap of the valve plus the lead, then take a point o on the link $d j$ at a distance from d equal to about $1\frac{1}{3}$ times $f p$, and draw $o n$, cutting $h i$ in q; then q will be the fulcrum of the lever $o n$, and will also be the middle of a curved guide in which the point q slides, or the pivot to which the suspension link r is attached, if such a link is used. The position of the point o thus obtained is a first approximation only, and should be shifted on the link $d j$ if later in the design it is found that a more correct action of the valve may thus be obtained.

15. The valve rod l may have any convenient length, depending on the design of the engine; but the suspension link r must have the same length. If a curved guide is used instead of the suspension link, the latter should have a radius equal to the length of the valve rod. The distances marked s and t represent the lap plus the lead at the head and crank ends, respectively, while u and v represent the respective port openings for the two ends. The diagram shows that the valve has moved from the central position a distance equal to the lap plus the lead when the crank is on its dead center. The angular motion of the link $o q n$, accompanied by the swinging of the point q in its arc, causes the port to open. When the engine is on its dead center, the center of the sliding block or the pin of the link r

coincides with the center *q*, and the curved guide may be tilted or the suspension link may be moved to any position without moving the valve. Hence, the lead must be constant for all positions of the reverse lever.

16. If it is desired to study the valve motion carefully for every position of the crankpin, this can be done through the harmonic diagram by simply laying off the displacements of the valve from the mid-position for various positions of the crankpin, as shown on the diagrams already explained. The Joy gear gives a much sharper cut-off at high ratios of expansion than the link motion, and an excellent steam distribution for all points of cut-off. In laying out this style of gear, however, the designer should observe carefully the release and compression, as at high ratios of expansion they are liable to be excessive.

THE CORLISS VALVE GEAR

17. Arrangement of Corliss Valves.—The Corliss valve arrangement consists of four cylindrical valves, *a*, *b*, *c*, and *d*, Fig. 15, *a* and *b* being the steam valves, and *c* and *d* being the exhaust valves. This illustration shows a typical arrangement of one half of a Corliss cylinder, and a diagram of the valve motion of the other half. The steam enters the steam chest *e* on top of the cylinder by the nozzle *f*, and exhausts from the exhaust chest *g* beneath the cylinder through the nozzle *h*. The valves receive an oscillating motion from a wristplate, indicated by the circle *n*, pivoted at the middle of one side of the cylinder. The exhaust valves *c* and *d* are opened and closed positively by the wristplate, while the steam valves *a* and *b* are opened by the wristplate and closed by dashpots.

Fig. 16 shows an excellent design of a double-ported Corliss valve, in which *a* and *b* are the two ports. The valve is divided into two portions *c* and *d* with a passage *e* between them through which the steam passes to one of the ports. Exhaust valves of this form are sometimes set partly in the

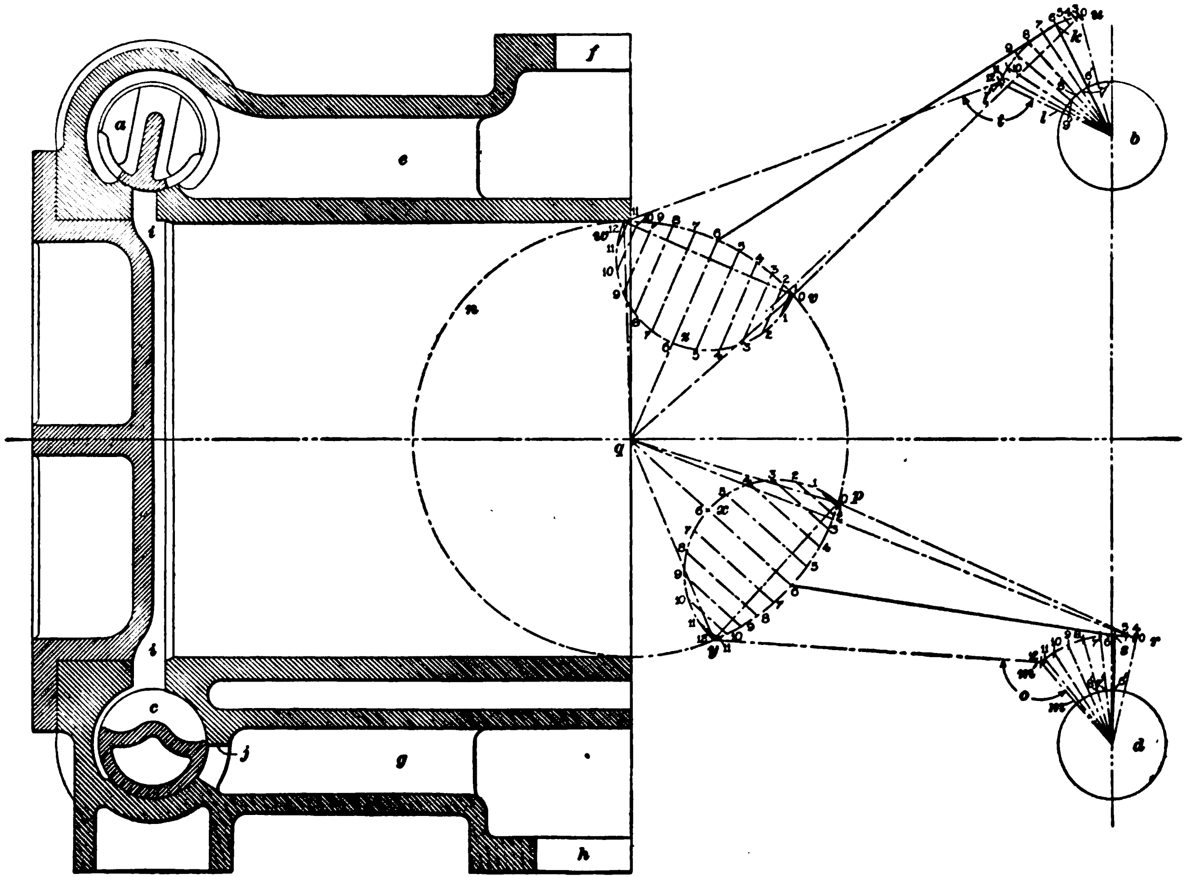


FIG. 15

cylinder, and the backs of the valves cut away to clear the piston. A considerable reduction in the clearance volume can be effected in this manner.

In Fig. 17 is shown a form of Corliss valve much used. This is virtually a trick valve, taking steam at two points, *a* and *b*. Owing to the large unbalanced area under this valve when closed, it requires a great deal of force to move it; it wears rapidly, and therefore should not be used with high steam pressures.

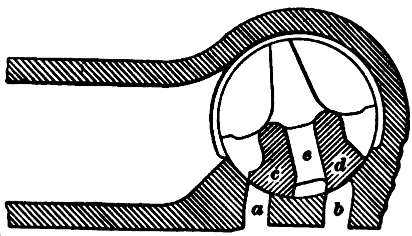


FIG. 16

18. Diameter of Corliss Valves.—The first step in the design of the Corliss valve gear is to determine whether the valves shall be single-ported or double-ported. Single-ported valves may be used for all ordinary speeds, and they are less expensive. When it is desired to operate at high speeds, it is desirable to keep the valves and valve gear as small and light as possible. This can best be accomplished by making the valves double-ported.

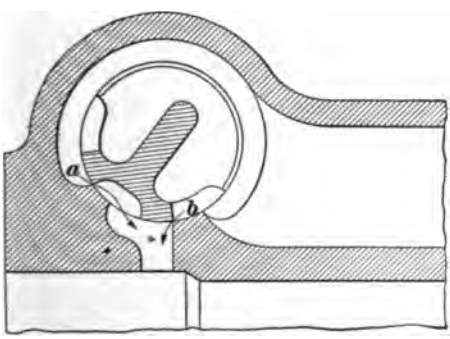


FIG. 17

The diameter of the valves for single ports is usually made one-fourth the diameter of the cylinder, and for double ports, one-fifth the diameter of the cylinder. In the case of double ports, great care should be taken to get the ports a sufficient distance apart. The bridge or seat between the ports should be 1.8 times the width of the ports. Care should also be taken that the angle included between the extreme outside edges of the two ports does not exceed 140°.

19. Ports and Passages.—The areas of the ports and passages of Corliss engines may be the same as in the case of slide-valve engines, as explained in *Valve Gears*, Part 1. The location of the steam ports relative to the piston when the latter is at the end of its stroke should be such that the entering steam will not strike the top of the piston; therefore, the steam port should be set back in the head. In practice, it is customary to locate the back edge of the exhaust port in line with the back edge of the steam port and make the recesses *i*, *i*, Fig. 15, in the cylinder head exactly the same for steam and exhaust in order that the cylinder head may be put on either side up. These recesses in the cylinder heads do not extend all the way around the head, but for the length of the port only. The edge *j* of the exhaust port should not be above the horizontal center line of the valve; if it is, leakage by wear will probably result.

20. Wristplate and Valve Arms.—The next points to be determined are the lengths of the steam arms *l*, exhaust arms *m*, the diameter of the wristplate circle *n*, and the location of the pins on the wristplate. The length of the exhaust arm *m* is generally made equal to the diameter of the valve, while the steam arm *l* is made about 15 per cent. longer than the exhaust arm. The extreme angular vibration of the wristplate should not exceed 50°. Since the exhaust port is wider than the steam port, and the valves are made of the same diameter, it follows that the angular motion of the exhaust arm *m* must be greater than the angular motion of the steam valve arm *l*, and the location of the wristplate pin for the exhaust arm should be determined first. This must be done tentatively, but usually it can be located definitely by the second trial.

In determining the angular motion of the exhaust valve, a lap of one-thirtieth the diameter of the valve should be given, and a similar amount of overtravel; the angle *o* should not exceed 140°. When the wristplate pin is in its extreme position *p*, it should pass about 8 per cent. of its travel beyond a straight line joining the center of the wristplate *q* with the valve-arm pin *r* when in its extreme position. The ratio of the length of the arc *rs* to *rm′* should be, approximately, as 1 is to 5, the point *s* being the valve-arm pin position corresponding to the mid-position of the wristplate pin *p*. When the radius of the wristplate and the location of the pin are found according to the above limitations, a satisfactory action of the exhaust valve will result.

21. In engines using a single wristplate, the wristplate pins for steam connection are usually, though not necessarily, located on the same radius as the exhaust wristplate pins, and the lengths of valve arms are given for this arrangement. The wristplate pins and the position of the steam-valve arms are located tentatively according to the following limitations: The angle *t* should not exceed 140°. When the valve-arm pin *u* and the wristplate pin *v* are in their extreme positions, the point *v* should travel past a line joining the center *q* of the wristplate with the center of the pin *u*, about 5 per cent. of the total travel of the wristplate. The ratio of the angular movement *ku* of the valve to the total angular movement of the pin *u* on the valve-arm pin circle should be as 1 to 4, *k* being the position of the valve-arm pin *u* corresponding to the mid-position of the wristplate. In determining the angular motion of the arm *l*, the valve should have a lap of about one twenty-fourth of its diameter and an overtravel equal to the lap. The dash pot should be connected at a radius of about 1.3 times the radius of the valve. If it is found impossible to secure the required valve action by the above directions, the designer must use his judgment in making alterations that will give the desired result.

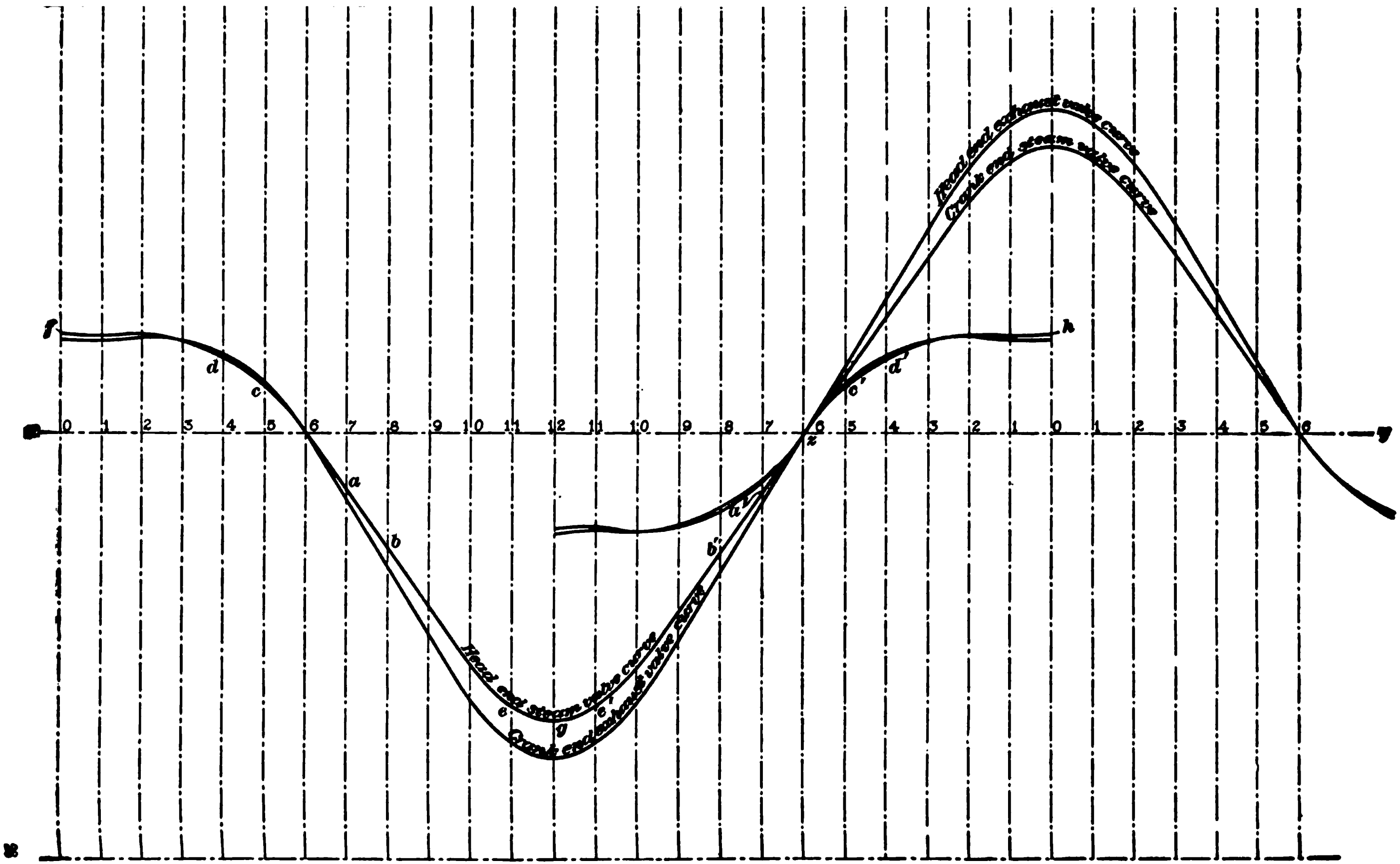
22. Harmonic Diagram for Corliss Valves.—Having determined the general proportions, arrangement, and motions of the Corliss gear, it is desirable to analyze the motions of the valves in relation to the motion of the piston. This is readily done by the aid of the harmonic valve diagram.

The first step is to lay out the piston-displacement curve, using the correct lengths of connecting-rod and crank, as explained in connection with the plain slide valve described in *Valve Gears*, Part 1. For convenience, the piston curve should be plotted for every 15° of motion of the crank, which will give twenty-four equal divisions on the diagram. This curve should be drawn clearly and carefully on a large sheet.

The next step is to draw the displacement curves of the valves. On the skeleton diagram of the valve motion in Fig. 15 lay off the arcs *p y* and *w v*, representing the travel of the wristplate pins. Since the angularity of the eccentric rod in the Corliss gear is small, its effect on the wristplate motion may be neglected. On the chords *p y* and *w v*, as diameters, draw the semicircles *p x y* and *w z v*. These semicircles then represent the motion of the eccentric center to the same scale as the chords *p y* and *w v* represent the throw of the eccentric. Divide each semicircle into twelve equal arcs, thus making the number of divisions in a half revolution of the eccentric equal to the number of divisions in a half revolution of the crank, as used in laying out the piston-displacement curve. Denote these points on the semicircles by the figures, *1*, *2*, *3*, etc., as indicated, and through them draw lines at right angles to the chords *p y* and *w v*. Extend these lines until they intersect the wristpin circle in the points *1*, *2*, *3*, etc., as shown. These latter points represent the positions of the wristplate pin at the corresponding positions of the eccentric, as indicated by the numbers on the semicircles. Then, with the lengths of the steam and exhaust valve rods *6 k* and *6 s* as radii, and with the points *1*, *2*, *3*, etc., as centers, locate the corresponding positions of the valve pins *u* and *r*, respectively, as shown on the arcs *u l'* and *r m'*.

23. The valve-displacement curves are now constructed as shown in Fig. 18, the following method being used: On a sheet of tracing paper lay off a straight line *x y*, and divide it into thirty or more equal parts, making the divisions equal in length to the divisions used in constructing the piston-displacement curve. The position *6 k* of the steam valve rod, Fig. 15, is the position corresponding to the mid-position of the wristplate. Consequently, on the tracing-paper diagram, Fig. 18, the point on the valve curve corresponding to the position *k* of the valve pin must lie on the line *xy*, since it is the position corresponding to a position of the eccentric 90° from the end of its throw, or mid-position. Hence, choose some point on *xy*, Fig. 18, near the left end, say the seventh point from *x*, and designate it by *6*. This point then represents the mid-position of the eccentric. It is customary to draw the valve-displacement curves full size, whereas Fig. 15 is drawn to a smaller scale. Hence, all measurements taken from Fig. 15 must be multiplied by the proper factor to give full-size dimensions before laying them off on Fig. 18.

When the eccentric has moved 15° from the mid-position as shown in Fig. 15, the steam-valve rod will occupy the position *7–7*, and the valve will have moved a distance represented by *6′–7′*, on the valve circumference. On Fig. 18, then, locate the point *7* on the line *xy*, 15° to the right of *6*, and on the vertical through that point lay off *7 a* equal to the chord *6′–7′*, Fig. 15, first multiplying the length *6′–7′* by the proper number to give the full-size displacement. Then *a* is another point on the head-end steam-valve curve. When the eccentric has moved another 15°, the pin *u* is at *8*, and the valve displacement is represented by *6′–8′*. So, on Fig. 18, make *8 b* equal to the full-size displacement represented by the chord *6′–8′*. The point *b* is then a third point on the curve. In a similar manner, locate the remainder of the valve displacements, from *6* to *12*, on the proper degree lines. The displacements *5 c*, *4 d*, etc. are laid off above *xy*, and are found from Fig. 15 in the same manner as were *7 a*, *8 b*, etc. Having located the points *a*, *b*, *c*, *d*, etc., a smooth curve is drawn through them, resulting in the curve *f 6 g*, which is the displacement curve of the head-end steam valve during its opening movement. Assuming that the releasing gear does not act, the displacement curve of this valve during its closing movement will be the same as that during opening, except that it will be laid off in reverse order. That is, *11 e′* is made equal to *11 e*, *8 b′* equal to *8 b*, *4 d′* equal to *4 d*, and so on, thus giving the closing half *g 6 h* of the curve. The complete curve *f 6 g 6 h* is, then, the displacement curve of



Head end exhaust valve curve
Crank end steam valve curve
Head end steam valve curve
Crank end exhaust valve curve
f
h
a
b
c
d
e
g
a'
b'
c'
d'
e'
z
y

FIG. 18

the head-end steam valve during one complete revolution of the engine.

24. While the steam valve at the head end is opening to admit steam, the exhaust valve at the crank end is opening to exhaust. That is, the same movement of the wristplate opens the head-end steam valve and the crank-end exhaust valve; consequently, the valve-displacement curves of these two valves lie side by side, as shown. The exhaust-valve curve is laid off in the same manner as was the steam-valve curve, except that the displacements are measured by the chords *6′–7′*, *6′–8′*, etc. on the valve circumference of the exhaust valve. Also, since the two steam valves and the two exhaust valves have like motions, the displacements for the valves at both ends of the cylinder may, without error, be taken from the head-end steam and exhaust valves, as shown in Fig. 15. The displacement curves for the crank-end steam valve and head-end exhaust valve are laid off, as shown in Fig. 18, by the method already described for the head-end steam valve, but above xy instead of below it, as in the previous case.

The point z, at which the crank-end steam-valve curve first cuts xy, must be 180° from the point *6* at which the head-end steam-valve curve first cuts xy, since the steam valves at opposite ends reach their corresponding displacements just 180°, or one-half a revolution, apart. Hence, the point z is located 180° to the right of the first point *6*, and this point z then becomes the starting point in drawing the displacement curves for the crank-end steam valve and head-end exhaust valve, just as the first point *6* was used as a starting point in drawing the displacement curves for the head-end steam valve and the crank-end exhaust valve.

In locating the several valve curves with reference to xy, the same method has been followed in the Corliss diagram as in the plain slide-valve diagram. That is, the curves representing the motions of the valves that open on the forward stroke are placed below the line xy, while the curves of the

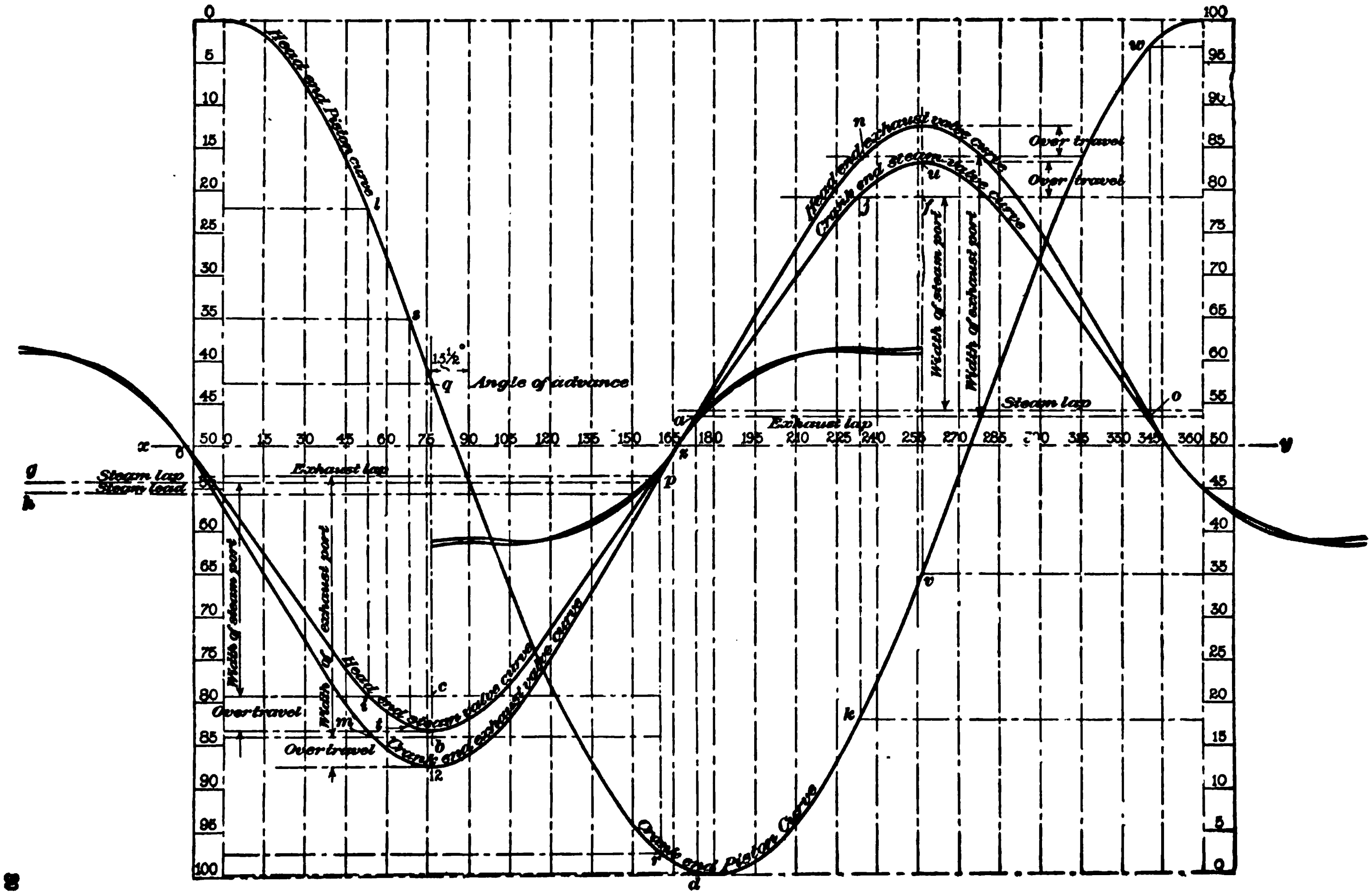

Head and Piston curve
Crank and Piston Curve
Head end exhaust valve curve
Crank end steam valve curve
Head end steam valve curve
Crank end exhaust valve curve
Angle of advance
15½°
Steam lap
Steam lead
Exhaust lap
Width of steam port
Width of exhaust port
Over travel

FIG. 19

valves that open on the return stroke of the piston are laid off above *x y*.

When the crank is in the position corresponding to the point *z*, where all four of the valve curves cross *x y*, the wristplate and the valves are all in mid-position. The peculiar flattened appearance of the extreme ends of the valve-displacement curves is due to the pause in the motion of the valves when the wristplate pins are in the positions corresponding to *p* and *v*, Fig. 15.

Having the valve curves now drawn on the tracing paper in their correct relative positions, the whole is placed over the piston curve so as to form the complete diagram shown in Fig. 19. To have the valve curves bear the proper relation to the piston curve, several points must be carefully noted. First, the lines *x y* of the two diagrams must coincide. Next, on the piston-curve diagram the line *g* is laid off below *x y* a distance equal to the lap, or about one twenty-fourth the diameter of the valve, and the distance *g h* is made equal to the lead, which may vary from $\frac{1}{32}$ to $\frac{3}{32}$ inch. Finally, a horizontal line is drawn from *h* until it intersects the vertical through the 0° position of the crank. Then the valve-curve tracing is shifted until the head-end steam-valve curve crosses this point of intersection of the lead line *h* and the vertical 0° line. The resulting diagram is that shown in Fig. 19.

25. The angle of advance is found by noting how far in advance of the 0° position of the crank the eccentric reaches its 90° position, or mid-position. Thus, the angle of advance is measured by the distance *6–0*, equal to $13\frac{1}{2}$°. It will be seen that the point *6* represents the 90° position of the eccentric, and the point *0* represents the dead-center position of the crank. The distance *6–0*, therefore, represents the angle the eccentric must travel beyond its 90° position before the piston reaches the end of its stroke, which is the angle of advance.

When the piston curve does not extend to the left beyond the 0° position of the crank, the angle of advance may be measured by noting the distance between the point at which the valve reaches its maximum displacement and the 90° position of the crank. In Fig. 19, the valve reaches its maximum displacement at *b*, and the distance between the vertical line *b q* and the 90° position of the crank is 13½°, as shown.

Having located the tracing of the valve curves in its correct position over the drawing of the piston-displacement curve, the displacement curves of the valves may be transferred to the drawing of the piston curve by pricking through, and a permanent record of the valve action may thus be obtained. The widths of the steam and exhaust ports are then laid off from their respective lap lines, above and below *x y*, as shown.

It will be observed that the maximum displacement of the head-end steam valve, as shown at *b*, is greater than the steam lap plus the width of the steam port by an amount *c b*. That is, the valve not only fully uncovers the port, when at its maximum displacement, but travels a distance *c b* beyond. This distance *c b* is called the overtravel. Similarly, the overtravel of the crank-end steam valve is represented by *f u*. In general, the overtravel is equal to the maximum displacement of a valve minus the sum of the port width and the lap. The amount of overtravel of each exhaust valve is clearly indicated in the figure. In the case of an exhaust valve, however, the sum of the exhaust lap and the width of the exhaust port is deducted from the maximum displacement to obtain the overtravel.

26. The full port openings occur at the points where the valve curves cross their respective outside port edges; that is, at *i* for the head-end and *j* for the crank-end steam valves, and at *m* for the crank-end and *n* for the head-end exhaust valves. By projecting the points *i* and *j* vertically to the piston curve at *l* and *k*, respectively, and then drawing horizontal lines from *l* and *k* to the percentage scales at the sides, it is found that full steam-port opening occurs at 22 per cent. on the forward stroke and at 18 per cent. on the return stroke.

The opening of the exhaust valves and the beginning of exhaust is represented by the points at which the curves of the exhaust valves in their opening movements cross their respective exhaust lap lines. This occurs at *a* for the head-end exhaust valve. Projecting *a* vertically to the piston curve at *d*, and then projecting *d* horizontally to the left percentage scale, it is seen that the exhaust valve opens at 99½ per cent. of the forward stroke, or ½ per cent. from the end of the stroke. A similar projection will show that exhaust takes place at 99½ per cent. of the return stroke. The points *p* and *o* where the exhaust valve curves, on the closing motion, cross their exhaust lap lines, represent the points of compression. Projecting these points to *r* and *w*, respectively, and thence to the percentage scales, it is seen that the exhaust valve closes at 97½ per cent. of the forward stroke and at 97 per cent. of the return stroke, giving 2½ and 3 per cent. compression, respectively, at the crank and head ends.

27. Cut-off by the action of the releasing gear must take place before or at the point of greatest displacement of the valve. An inspection of the diagram, Fig. 19, will show that the greatest displacement occurs when the piston occupies the positions denoted by the points *q* and *v*; projecting the points to the percentage scales at the sides, it is found that the latest point at which cut-off can occur is 42½ per cent. of the stroke for the head end, and 35 per cent. for the crank end. If the knock-off cams are set for equal cut-off at the early point, the cut-off will be unequal by the above amount at the latest points. This can readily be overcome by making the head-end cam-arm longer than that on the crank end. Since cut-off takes place later on the head end than on the crank end, and it is desired to have equal cut-off, the head-end knock-off cam must stand at earlier positions than the crank-end cam for the later points of cut-off.

In setting Corliss valves, the wristplate is placed in mid-position, the governor blocked up to approximately its normal running position, and the governor rods adjusted so as to give equal cut-off at any desired point. An engine of ideal design should then give equal cut-off for any other load; that is, the cut-off should be the same on both ends of the cylinder for any points from the mid-position of the valves to the latest point of cut-off. Assuming that the knock-off cams are adjusted to knock off when the valves are in mid-position, that is, the position corresponding to the 90° position of the eccentric, or the position indicated by the line xy, Fig. 19; then, if the cam-arms are equal, the cut-off will take place at 42½ per cent. on the head end and at 35 per cent. on the crank end, as indicated above. In order to cause cut-off to take place at 35 per cent. on the head end also, the head-end cam must lag somewhat behind the crank-end cam in its travel from the earliest to the latest positions, the amount of lag increasing gradually from the mid-position of the valve to the 35-per-cent. cut-off position. Thus, the angle through which the head-end cam must travel in its movement through the entire range of cut-off must be less than that of the crank-end cam, and in order that this may be so the cam-arm of the head end must be made longer than the cam-arm on the crank end.

28. The amount by which the head-end cam-arm must be lengthened is determined as follows: The latest point of cut-off for the crank end is 35 per cent., and it is desired to have the same cut-off on the head end; hence, projecting the 35-per-cent. point of the head end, Fig. 19, over to the piston curve, and thence down to the valve curve at t, the line st is obtained for the head end similar to the line uv for the crank end. To find the displacement of the valves at the maximum cut-off of 35 per cent., measure along the lines st and uv the distances from the line xy to the intersections with the valve curves. These measurements are found to be $1\frac{15}{32}$ inches for the head end and $1\frac{1}{4}$ inches for the crank end. These distances are the lengths of chords showing the valve displacement measured on the valve circumferences, Fig. 15.

If the knock-off cams are set for equal cut-offs at the beginning of the stroke, the motions of the cams must be proportional to the valve displacements up to the latest points of cut-off. From Fig. 19, it is seen that the valve motions, to give equal maximum cut-offs, must be $1\frac{15}{32}$ and $1\frac{1}{2}$ inches. But, if the cam-arms are equal, the motion of the head-end valve is $1\frac{1}{2}$ inches. Therefore, the head-end cam-arm must be lengthened by the ratio $1\frac{1}{2}$ to $1\frac{15}{32}$ in order to cause cut-off to take place when the head-end valve has a displacement of $1\frac{15}{32}$ inches. That is, if the lengths of the cam-arms to which the governor rods are attached are in inverse ratio to the lengths of these chords, they will produce equal cut-offs at both ends of the cylinder. Therefore, whatever length be chosen for the length of the crank-end cam-arm, the head-end cam-arm in the case considered will be lengthened in the proportion of $1\frac{1}{2}$ to $1\frac{15}{32}$, and the result will be equal cut-off, but at a sacrifice of some range at the head end. This same result may be obtained by choosing a length for the head-end cam-arm and shortening the crank-end cam-arm in the proportion of $1\frac{15}{32}$ to $1\frac{1}{2}$.

It will be seen from the preceding that in the single-eccentric Corliss valve gear the range of cut-off is limited, and even with the range of cut-off given there is very little compression, and the release will be late, in practice. The only way to secure more compression will be to move the eccentric ahead and increase the exhaust lap correspondingly, which will, however, reduce the range of cut-off still more.

29. Increased range of expansion can be obtained by providing separate eccentrics for the steam and exhaust valves and setting the steam eccentric back sufficiently, so that it will not reach its extreme throw until the piston has made about three-fourths of its stroke. In this arrangement the knock-off cams must be set to trip when the wrist-plate reaches its extreme throw with the governor at rest, in order that the releasing gear and dashpots may be in operation at all times. With the single-eccentric gear, the latch of the disengaging hook just touches the knock-off cam when the wristplate is in the extreme position and the governor is down, and cut-off begins as the governor begins to rise. With the double-eccentric gear, the wristplate for the steam valves may be omitted and the eccentric rods connected to the valve arms; in that case, double-ported valves are desirable. The reason is that when the eccentric is set for $\frac{3}{4}$ cut-off, both steam valves are open about half way, while the wristplate is in its central position. As has been seen, the first half of the motion of the wristplate produces very little motion of the valve; consequently, the valve is opened with a slower motion, and therefore it is desirable to use a double-ported valve, which reduces the motion of the valve. It is generally preferable to retain the wristplate motion in connection with the exhaust valves.

In laying out the valve-gear diagram with double-eccentric gear, the same procedure is followed as with single-eccentric gear, care being taken to properly locate the valve curves. The steam and exhaust curves will not in that case coincide at the point where they cross the line xy, and they will reach maximum travel at different times. The exhaust-valve curve should be moved ahead to get more compression, and the steam-valve curve should be moved back until the angle of advance becomes negative. This negative angle may, in some cases, be as great as 15° or 20°, thus making the angle that the steam eccentric radius makes with the crank as small as 70° or 75°.

OTHER VALVE GEARS

30. Poppet Valves.—Poppet valves have been employed for many years as steam-distributing valves, but only recently have circumstances demanded their use. With highly superheated steam, it is difficult, if not quite impossible, to lubricate satisfactorily any form of sliding valve; hence, it is necessary to use a type of valve that does not slide on its seat, which condition the poppet valve fulfils.

Figs. 20 and 21 show recent designs of inlet and outlet poppet valves for a large vertical engine using highly superheated steam. Fig. 20 shows the lower admission valve, which has four seats *a*, on the cage *b*, which is secured

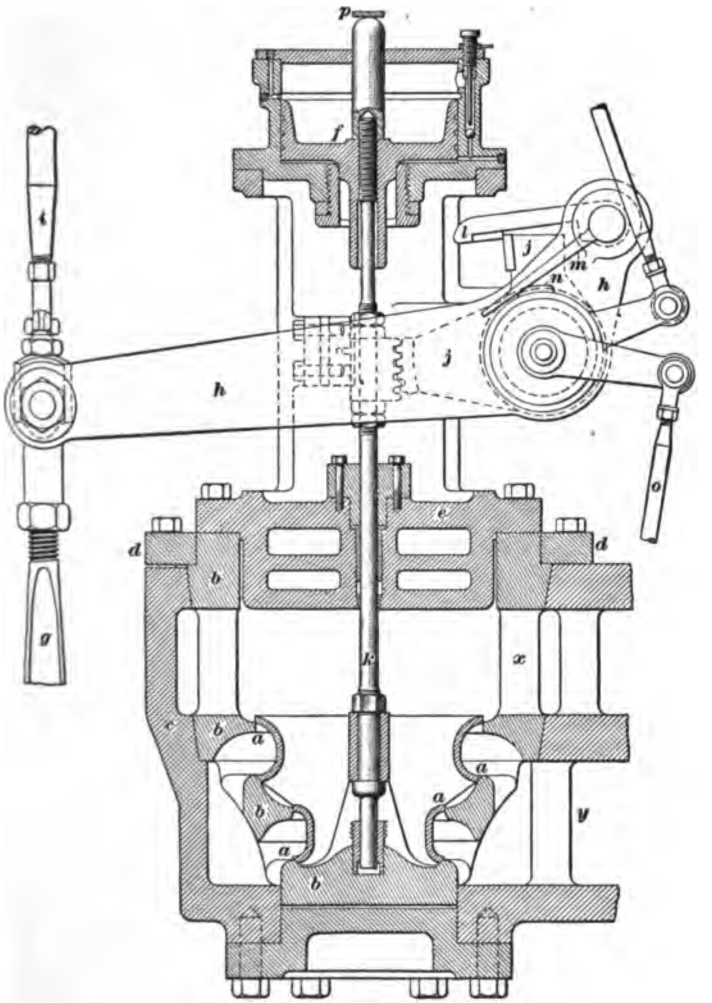


Fig. 20

to the valve body *c* by a heavy steel ring *d*. The bonnet *e* is bolted to the top of the cage *b* and carries the controlling mechanism and the dashpot *f*.

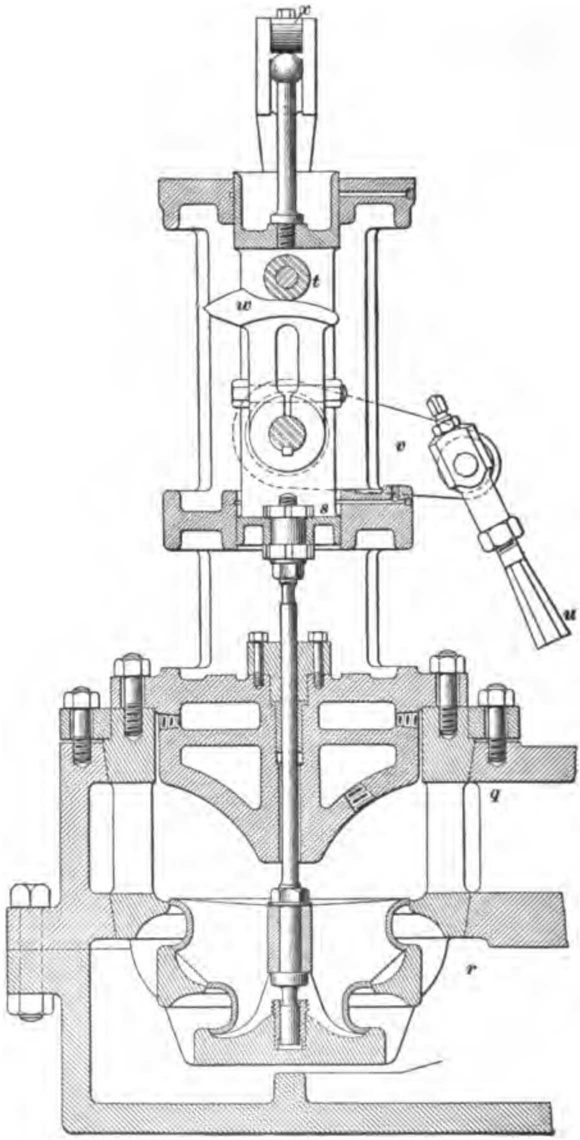


Fig. 21

The eccentric rod *g* engages with the valve arm *h* and the rod *i*, which communicates the motion of the eccentric to the valve arm of the upper valve mechanism. The bell-crank *j*, shown partly in dotted lines, is journaled loosely on the same pin as *h*. The horizontal arm of *j* is a segment of a gear that engages a section of rack attached to the valve stem *k*. The vertical arm of *j* is fitted with the usual steel block, with which a lifting hook *l* engages. This hook releases the bell-crank *j* when the tongue *m* strikes a knock-off cam *n*, the position of which is controlled by means of the governor rods *o*.

When the valve is seated, the steam pressure is above and inside of it. With the parts in the position shown in the drawing, the valve is just ready to open. The hook *l* is engaged with the steel block on the upper arm of *j*, and consequently as *g* moves up the arm *h* carries the bell-crank *j* with it, lifting the valve and deflecting the flat-leaf spring *p*. At the point fixed by the governor, the knock-off cam unlatches the hook *l*, releasing *j* from *h*, and the spring *p* closes the valve sharply. The motion, however, is checked by the dashpot *f* at the instant the valve is seating. The steam enters the valve through the port *x* and passes to the cylinder through the port *y*.

31. Fig. 21 shows the exhaust valve. The valve and cage are similar to the corresponding parts of the admission valve shown in Fig. 20. The arrangement differs only in that the port *q* is in communication with the cylinder while the port *r* conveys the exhaust from the valve. A guide *s* for the valve stem carries a pin with a hardened-steel roller *t*. The eccentric rod *u* engages with the exhaust arm *v*, to which the cam *w* is keyed. As *u* moves downwards, drawing the arm *v* with it, the cam *w* swings to the right, and the curved edge, passing under the roller *t*, raises the valve. The outline of the cam is such that the valve is opened quickly and is held stationary until the time for closing. A flat spring *x* holds the roller *t* in contact with the cam *w* at all times, so that the motion of the valve is smooth and steady.

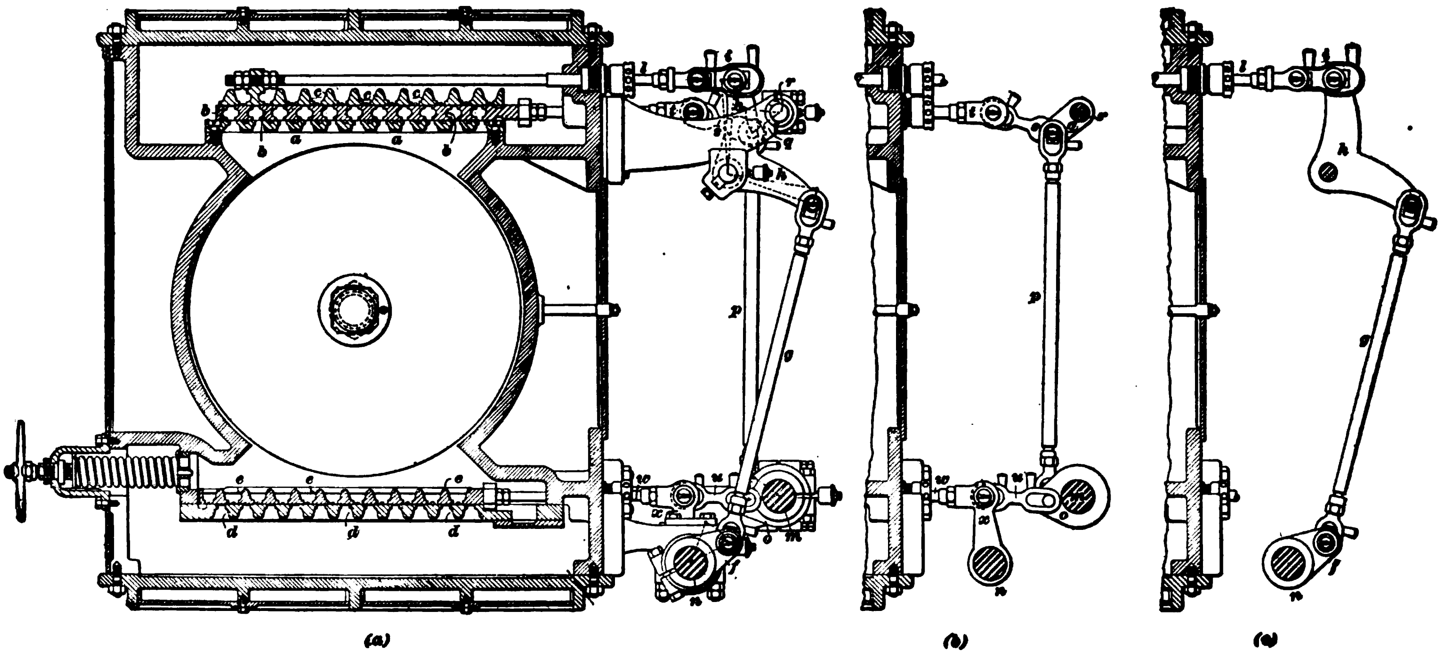


Fig. 22

In some recent pumping-engine designs, single-seat poppet valves have been used with the object of reducing the clearance to a minimum. The force required to lift a single-seat poppet valve from its seat would be very great were it not for the fact that the compression is properly adjusted to balance the valve at the instant of opening.

32. Gridiron Valves.—Fig. 22 is a cross-section through the ports and valves of one end of the cylinder of a McIntosh & Seymour medium-speed engine. The steam valve seat consists of a number of narrow bars *a*, *a* with spaces between them for the passage of steam to the port. The steam valve is a frame with a similar series of bars *b*, *b*, having passages between them corresponding to the passages between the bars of the seat. To open the ports, the valve is moved so as to bring the passages between its bars over those between the bars of the seat.

A cut-off valve is made up of a frame with bars *c*, *c*. It slides on the back of the main steam valve and cuts off the steam supply by closing the passages through the main valve. The exhaust-valve seat and valve are, respectively, made up of bars *d*, *d* and *e*, *e* similar to the steam-valve seat and valve.

33. Valves of this type are called **gridiron valves** from the resemblance they have to a gridiron. Their advantage lies in the fact that a liberal port opening can be obtained with a short range of travel; also, the wear of the valve and seat and the power required to operate the valve are considerably reduced. But the resistance to the flow of steam through the narrow passages between the bars is considerable, and the aggregate area of the passages must therefore be greater than the area that would be required with a valve like a plain slide valve, which provides for the passage of all the steam through a single large opening. By the use of a sufficiently large number of openings, however, the total area can easily be made great enough to prevent any serious loss from wiredrawing.

34. In the McIntosh & Seymour engine, the valves are driven by two eccentrics—one, which may be called the main eccentric, for the steam and exhaust valves, and the other for the cut-off valves. The main eccentric is keyed to the engine shaft, and by means of a short rod transmits an oscillating motion to an arm keyed to the shaft *m*, Fig. 22 (*a*), which extends along the side of the engine bed and cylinder. Two cranks *o* are connected to the shaft *m*, one for each end of the cylinder. Each crank transmits motion to the steam valve and exhaust valve at the end where it is located. The motion of the rocker-shaft *m* is transmitted to the steam valve of the end of the cylinder shown in Fig. 22 (*a*), through the arm *o* and rod *p*, and the togglejoint formed by the arm *q* pivoted at *r* and the link *s* connected to the end of the valve stem *t*. To show the relation between these parts more clearly, they have been drawn separately in Fig. 22 (*b*), where they are given the same letters as in Fig. 22 (*a*). A togglejoint connection to the exhaust-valve stem *w* is formed by the arm *o* and the link *u*. The arm *x* is pivoted loosely on the shaft *n* and serves as a support or guide for the joint between the end of the valve stem *w* and the link *u*. A similar set of connections serves to drive the steam and exhaust valves for the other end of the cylinder.

35. A study of the arrangement of the parts illustrated in Fig. 22 (*b*) shows that the nearer the links *q* and *s* are to being in line with each other, the less motion will be imparted to the valve by a given angular motion of the shaft *m*. When one of the togglejoints—the one driving the steam valve, for example—is in the angular position, the links of the other lie in a straight line. The straight-line position of each of the togglejoints corresponds to the period during which its valve is closed; the valve, therefore, has but little motion when the ports are closed. As the joint leaves the straight-line position, the motion of the valve becomes more rapid; and when it has moved through a distance equal to the lap, it has acquired considerable speed, so that at the point of opening it moves quite rapidly. Thus the valve has a rapid motion during the period of opening and closing, with a period of nearly complete rest during the time when it remains closed.

The effect of the togglejoints is modified by the action of the eccentric and its connection to the oscillating shaft *m*. By comparing the motion due to the togglejoint with the wristplate motion of the Corliss engine, it will be found that the effects on the motions of the valves are quite similar.

36. The motion of the cut-off valve, for the end of the cylinder shown in Fig. 22 (*a*), is imparted to it by a separate cut-off eccentric through an oscillating shaft *n*, the rocker-arm *f*, the rod *g*, the bell-crank *h*, the link *i*, and the valve stem *l*. These parts are also shown in Fig. 22 (*c*). A similar set of connections transmit motion from the shaft *n* to the cut-off valve on the other end of the cylinder. The point in the stroke at which the cut-off valve covers the passages through the main steam valve, and so cuts off the supply of steam to the cylinder, is varied by varying the angle of advance of the eccentric by means of a shaft governor.

MECHANICS OF THE STEAM ENGINE

CRANK-EFFORT DIAGRAMS

1. Tangential Pressure.—The pressure on the piston of an engine is transmitted to the crankpin through the piston rod and connecting-rod. The pressure on the crankpin may then be resolved into two principal components. One of these acts radially along the crank and exerts a pressure on the main bearing. The other, known as the **tangential pressure**, acts at right angles to the crank. It is evident, therefore, that this tangential pressure causes the crank to revolve; consequently, it is termed the **crank-effort.**

2. In Fig. 1, let OC represent the crank; then the circle $C'\,C''\,M$ represents the path of the crankpin. The steam exerts, through the piston, piston rod, and connecting-rod, a pressure on the crankpin C. Let it be assumed that the connecting-rod is infinitely long, so that the direction of the pressure on the pin is always horizontal. When the crank is in the position OC', the horizontal pressure of the steam simply produces a pressure on the bearing of the crank-shaft;

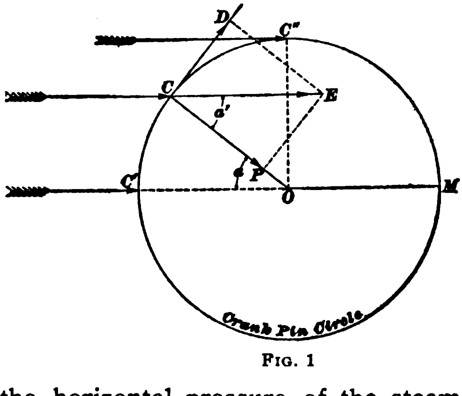


FIG. 1

there is no tendency whatever to turn the crank around O as a center; therefore, it is said to be on the dead center. When the crank is in the position $O\,C''$, all the pressure of the steam on the piston is expended in turning the crank around O as a center, and there is no radial pressure on the bearing. This is because the direction of the pressure at this point is at right angles to the crank, or, in other words, tangent to the crank-pin circle.

When the crank is in some other position, as $O\,C$, there will be a tendency to turn the crank around O as a center, and also to produce a pressure on the crank-shaft bearing. To find the magnitudes of the forces tending to rotate the crank and to produce a pressure on the bearing, let $C\,E$ represent to some scale the horizontal pressure or force on the crankpin at C. The turning force acts in the direction of the tangent $C\,D$, while the force that produces the pressure on the bearing acts along the crank, or in the direction $C\,O$. The force $C\,E$ may be resolved in the two directions $C\,D$ and $C\,O$ by means of the parallelogram of forces. From E, draw $E\,P$ parallel to $C\,D$ and $E\,D$ parallel to $C\,O$. Then, $C\,D$ is the tangential or turning force, and $C\,P$ the force producing pressure on the bearing, both to the same scale as $C\,E$.

Let a be the angle $C\,O\,C'$ that the crank makes with the horizontal. $C\,E$ and $C'\,O$ are parallel, and $C\,P$ and $C\,O$ coincide; hence, by geometry, $C\,O\,C'$ and $P\,C\,E$ are equal, or angle $a = a'$.

Tangential force $= C\,D = E\,P = C\,E \sin E\,C\,O = C\,E \sin a$, or *the force tending to turn the crank is equal to the horizontal force on the crankpin multiplied by the sine of the angle that the crank makes with the horizontal.*

When the crank is at $C'\,O$, a is zero; therefore, $\sin a$ is zero and the tangential force is zero, as it should be. When the crank is at $O\,C''$, a is 90°, $\sin a$ is 1, and the tangential force is the same as the horizontal force.

The radial force, or the force that exerts pressure on the bearing, may be shown in the same manner to be equal to the horizontal force multiplied by the cosine of the angle which the crank makes with the horizontal; thus,

$$\text{Radial force} = C\,P = C\,E \cos E\,C\,O = C\,E \cos a$$

EXAMPLE.—The horizontal force on the crankpin is 6,000 pounds; what will be the tangential and radial forces when the crank makes an angle of 60° with horizontal direction of the force?

SOLUTION.— $a = 60°$; $\sin a = .866$; $\cos a = .5$.

Tangential force,

$$6{,}000 \times \sin a = 6{,}000 \times .866 = 5{,}196 \text{ lb. Ans.}$$

Radial force,

$$6{,}000 \times \cos a = 6{,}000 \times .5 = 3{,}000 \text{ lb. Ans.}$$

3. A diagram showing the tangential pressure for every

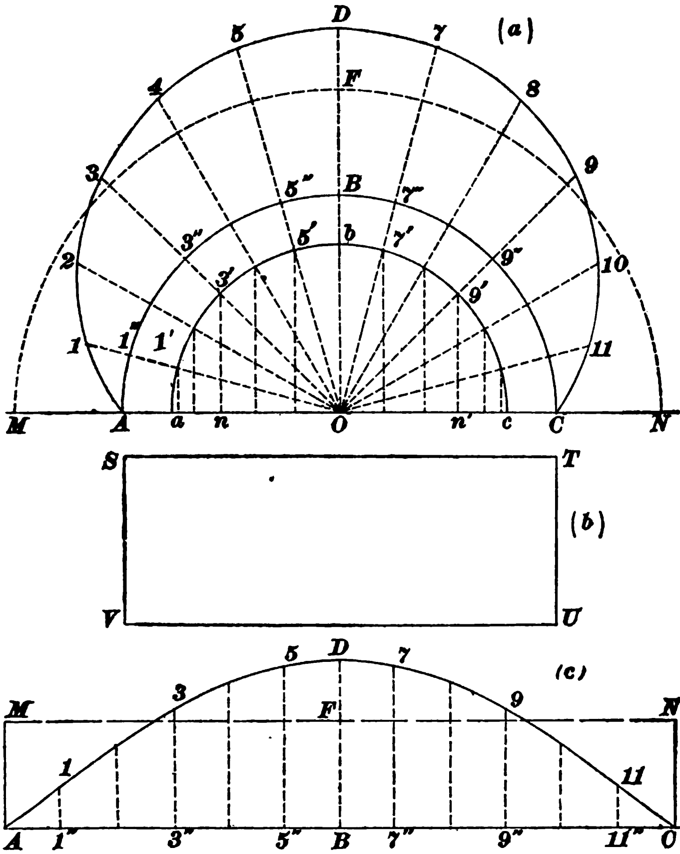


FIG. 2

position of the crank may be easily constructed, as shown in Fig. 2. For simplicity, assume the piston pressure to be constant throughout the stroke, as shown by the indicator diagram, Fig. 2 (*b*); suppose, also, that the connecting-rod is infinitely long, so that the direction of pressure is always horizontal. Let the length of the crank be OA, Fig. 2 (*a*); then, with O as a center and OA as a radius, describe one-half the crankpin circle ABC. Let Oa represent the uniform pressure SV, Fig. 2 (*b*), and describe the semicircle abc, Fig. 2 (*a*). Divide abc or ABC into a convenient number of equal parts (12 in this instance), and through the points of division draw the radial lines $O1$, $O2$, $O3$, etc., prolonging them beyond ABC. From the points where these radial lines intersect the semicircle abc, drop perpendiculars on the line ac; each perpendicular represents the tangential component of the pressure when the crank is in that position. For example, when the crank is at $O3'$, the length $3'n$ represents the tangential pressure, since $3'n = O3' \sin 3'On = Oa \sin 3'On$ = horizontal pressure × sine of the crank angle.

Now, lay off these perpendiculars, each on its own radial line outwards from the crankpin circle ABC; that is, lay off $3'n$ on the radial line $O3$, the length $3''–3$ being made equal to the length $3'n$. A series of points $1, 2, 3$, etc. is thus obtained; the curve ADC, drawn through these points, will represent the tangential pressures for all points of the stroke.

It will be noticed that at A and C, the dead centers, the tangential pressure is 0; and at D, it is equal to the horizontal pressure.

In Fig 2. (*c*) the crank-effort diagram, or tangential-pressure diagram, is represented with a straight base. The semicircle ABC has been straightened out, the ordinates $1''–1$, $2''–2$, etc., remaining the same as before.

Since the ordinates of Fig. 2 (*c*) represent the tangential pressures on the crank to the same scale that the ordinates of Fig. 2 (*b*) represent pressures on the piston, and since the length AC represents the distance passed through by the crank to the same scale that VU represents the distance passed through by the piston, it follows that the area of (*c*) represents the work done by the crankpin during a half revolution of the crank, or during one stroke of the piston.

The work done on the piston by the steam must be equal to the work given up by the crankpin. Therefore, since (*b*) and (*c*) have the same scale of pressures and distances, since (*b*) represents the work done on the piston and (*c*) represents the work done on the crankpin, their areas must be equal; and so they will be found to be by actual measurement.

The mean ordinate of (*c*) may now be found from the above considerations. The length of the semicircle ABC, Fig. 2 (*a*), is $\frac{\pi}{2}$ times the length of the diameter AC; but the base ABC of the diagram (*c*) is equal in length to the semicircle ABC, and the length VU of (*b*) is equal to the diameter AC, since both represent the length of stroke. Therefore, the base ABC of (*c*) is $\frac{\pi}{2}$ times as long as the base UV of (*b*). The areas of (*b*) and (*c*) are the same; consequently, the mean ordinate of (*b*) must be $\frac{\pi}{2}$ times that of (*c*), or $SV = \frac{\pi}{2} AM$; therefore, $AM = \frac{2\,SV}{\pi}$.

That is, *the mean ordinate of the diagram of tangential pressures on the crankpin, generally called the crank-effort diagram, is always* $\frac{2}{\pi}$ *times the mean ordinate of the diagram from which the crank-effort diagram is constructed.*

In both (*a*) and (*c*), MFN is the line of average tangential pressures, and is drawn in both cases parallel to ABC at a distance from it equal to the mean ordinate AM.

4. Tangential Pressure With Oblique Connecting-Rod.—The simple method of finding the tangential pressure just described is not applicable to a case in which the connecting-rod is of finite length. Instead of being horizontal at all positions of the crank, the connecting-rod stands at an angle to the center line of the engine, except when on the dead center. For such a case as this, the following graphic method will give accurate results:

Let OC, Fig. 3, represent the crank in any position, and BC the connecting-rod; also, let OC represent, by its length, the net forward pressure per square inch of piston area at the position B of the crosshead. Through the crank-center O draw the line MN perpendicular to the line of stroke OB and prolong BC, the center line of the connecting-

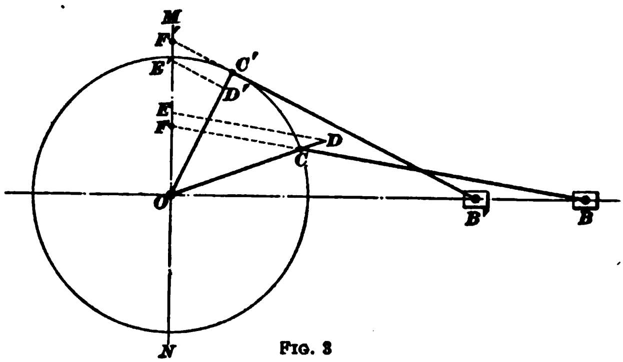


FIG. 3

rod, until it intersects MN in the point F. Let P and T denote, respectively, the pressure on the piston and the tangential component of the pressure on the crankpin; then,

$$T : P = OF : OC$$

or

$$T = P \times \frac{OF}{OC}$$

Expressed in words, the intercept OF represents the tangential pressure on the crankpin to the same scale that

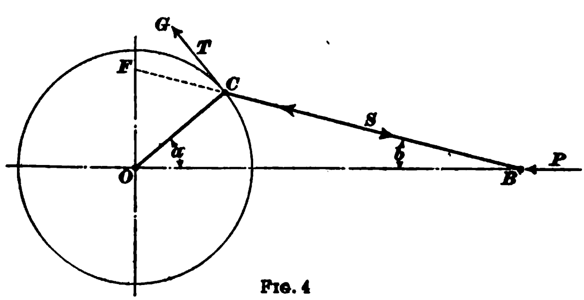


FIG. 4

the crank-radius OC represents the pressure on the piston.

For another position of the crank, as OC', the connecting-rod has the position $B'C'$, and the intercept OF' represents the tangential pressure on the crankpin.

The proof of this construction depends on the principle of resolution of forces. In Fig. 4, let P be the pressure on the piston; S, the pressure of the connecting-rod on the crankpin, and also the equal pressure of the rod on the crosshead pin at B; and T, the tangential component of the force S acting on the crankpin at C. It is evident that for the crosshead pin at B to be in equilibrium, the component of the force S in the direction OB must be just equal to the force P. The angle that the connecting-rod makes with the center line OB is $OBC = b$; hence, the component of S along the line OB is $S \cos b$; therefore,

$$P = S \cos b \tag{1}$$

The pressure S against the pin C is resolved into two components: one, the tangential component T in the direction CG; and the other, the radial component in the direction of the radius OC. By this resolution, $T = S \cos GCF$. Referring to the triangles OBF and OCF, it is evident that since FOB is a right angle, angle $OFC = 90° - b$; also, angle $OCF = a + b$, and angle $GCF = 90° - OCF = 90° - (a + b)$; further, angle $COF = 90° - a$. Substituting the value of angle GCF in the expression for T and remembering that the cosine of the complement of an angle is equal to the sine of the angle,

$$\begin{aligned} T = S \cos GCF &= S \cos [90° - (a + b)] \\ &= S \sin (a + b) \end{aligned} \tag{2}$$

Dividing equation (2) by equation (1),

$$\frac{T}{P} = \frac{S \sin (a + b)}{S \cos b} = \frac{\sin (a + b)}{\cos b} \tag{3}$$

In the triangle OCF,

$$\frac{OF}{OC} = \frac{\sin OCF}{\sin OFC} = \frac{\sin (a + b)}{\sin (90° - b)} = \frac{\sin (a + b)}{\cos b} \tag{4}$$

Combining equations (3) and (4),

$$\frac{T}{P} = \frac{OF}{OC}, \text{ or } T : P = OF : OC$$

the relation to be proved.

5. Net Forward Pressure.—In an engine two pressures exist in the cylinder at any instant, one on each side of the piston. The pressure in the working end, forcing the piston forwards, is called the **forward pressure**. That in the other end, resisting the forward motion of the piston, is called the **back pressure**. Hence, the difference between these two forces at any instant gives the **net forward pressure** at that instant. This pressure may easily be obtained from the indicator diagrams, for any given position of the piston, as shown in Fig. 5. Let OX represent the length of stroke of the piston, and OP the admission or boiler pressure of the steam. Then, when the piston has moved from O to a, the net forward pressure on the piston is obtained by erecting an ordinate ab and taking the intercept $a'b$, which is the difference between the forward pressure ab and the back

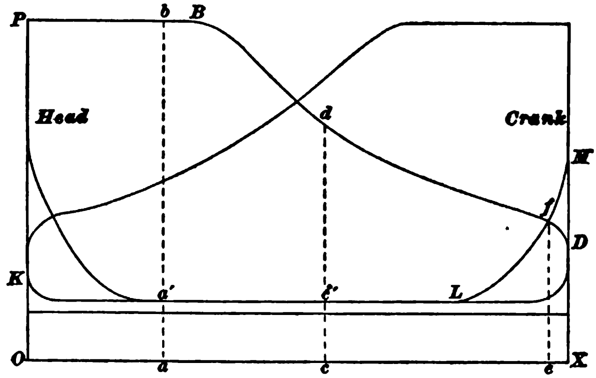


FIG. 5

pressure aa'. This length, in inches, multiplied by the scale of the spring, or the pounds per inch of the vertical height of the diagram, will give the net forward steam pressure at that point of the stroke. Similarly, the pressure may be obtained at other positions, as cd or ef. It will be noticed that at ef, the back pressure is equal to the forward pressure, and the net forward pressure is therefore zero.

It must be understood that the back pressure of the head-end diagram must be subtracted from the forward pressure of the crank-end diagram, point by point. It would be wrong to take as the net forward pressure the difference between the forward-pressure and the back-pressure lines of the same diagram, since each diagram represents the forward and back pressures on the same side of the piston, which occur on successive strokes. In order to obtain the net forward pressure, therefore, it is necessary to take the difference between the forward pressure of one diagram and the back pressure for the same point of the stroke of the diagram taken from the opposite end of the cylinder.

6. Diagram of Net Forward Pressures.—Since the net forward pressure is found for any piston position by measuring the ordinate included between the forward-pressure line of one diagram and the back-pressure line of the opposite diagram, it is evident that a diagram may be drawn that will represent correctly the varying net forward pressure

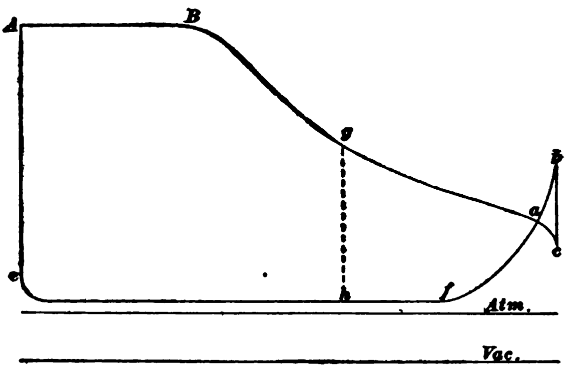


FIG. 6

during an entire stroke. This is easily done by combining into one diagram the two curves $PBfD$ and $KLfM$, Fig. 5, the former being the line of forward pressure on the forward stroke, and the latter being the line of back pressure on the same stroke. The combined diagram is shown in Fig. 6, in which $ABac$ represents $PBfD$, and $efab$ represents $KLfM$. Joining A and e, and b and c by straight lines, the figure is completed.

Any ordinate, as gh, measured to the scale of the indicator spring, is the net pressure on the piston urging it forwards when it occupies the position h of its stroke. The net pressure is 0 at a; that is, the pressure is the same on both sides of the piston. Between a and bc the net pressure is negative, or, in other words, the back pressure is greater than the forward pressure, and the piston is carried to the end of its stroke solely by the energy stored in the flywheel.

The ordinates of Fig. 6 have been laid off in Fig. 7 on a

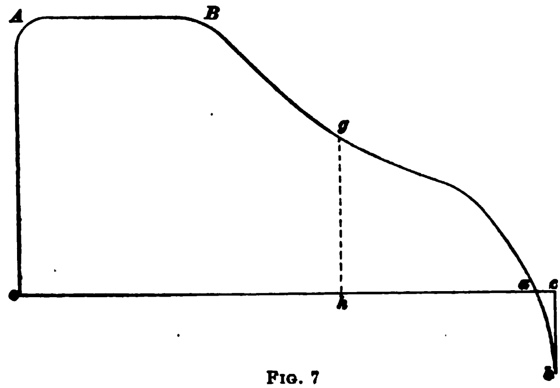


FIG. 7

straight base line ec, which is the line of zero net pressure, and show a little more clearly the net forward pressures on the piston for the different positions of the stroke. From a to c, it is quite apparent that the pressures are negative, because the positive ordinates are above and the negative ordinates below, the base line ec.

7. Inertia Pressure of Reciprocating Parts.—The combined diagram, Fig. 7, would be the outline from which to construct the crank-effort diagram, provided that the net pressures on the piston were the only forces acting. But this is not the case. At the beginning of each stroke, the reciprocating parts, which have considerable weight, must be started from a state of rest, accelerated to a maximum speed near the middle of the stroke, and then brought to rest again at the end of the stroke. Since the inertia of a body always opposes any increase or decrease in the volocity of that body, it is evident that the inertia of the reciprocating parts must exert an influence that may very properly be called **inertia pressure.**

During the first portion of the stroke, while the velocity of the reciprocating parts is increasing, the inertia of these parts opposes the forward pressure of the steam. Hence, the actual pressure exerted on the crosshead pin at any point in the first part of the stroke is equal to the difference between the net forward pressure and the inertia pressure at that point.

During the latter part of the stroke, the velocity of the reciprocating parts is decreasing, and consequently their inertia is exerting a pressure in the same direction as the forward pressure of the steam. Therefore, during this part of the stroke, the actual pressure on the crosshead pin at any point is equal to the sum of the net forward pressure and the inertia pressure at that point.

8. Inertia Pressure With an Infinitely Long Connecting-Rod.—The acceleration of the reciprocating parts at all positions of the crank may be determined, and the inertia pressure at any point may then be obtained by multiplying the acceleration by the weight of the reciprocating parts and dividing the product by 32.16. It has, however, been found that the same results are obtained in a shorter and simpler manner by considering the whole weight of the reciprocating parts as being concentrated at the crankpin center and as revolving at a uniform velocity. The inertia pressure at any instant is then exactly equal to the horizontal component of the centrifugal force due to the revolving weight at the crankpin.

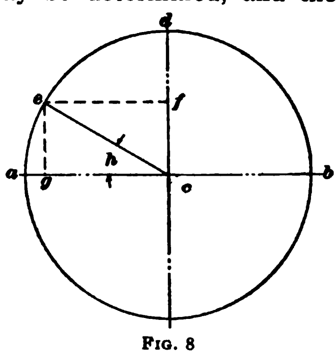


FIG. 8

In Fig. 8, let *a b* represent the stroke of an engine with an infinitely long connecting-rod and let *a d b* represent the crankpin circle. Assume that the crank *e c* has turned through the angle *h* and that the weight of the reciprocating parts is concentrated at *e*. Evidently, the rotation of the crank will set up a centrifugal force in the weight at *e*, acting radially outwards in the direction of *c e*. Let the radius *c e* represent, to some scale, this centrifugal force. Then this force may be resolved into two components, one horizontal, as cg, and the other vertical, as cf.

The vertical component cf, acting at right angles to the line of motion of the piston, can therefore have no effect on its motion. The only force that can influence the to-and-fro motion of the piston is that which acts parallel to the line of motion of the piston; that is, the horizontal component cg. In other words, the horizontal component of the centrifugal force at any crank position represents the inertia pressure of the reciprocating parts at that position. This pressure may be found by dropping a perpendicular from the end of the crank to the line ab. The distance from the center c to the foot of this perpendicular will then represent the inertia pressure for the crank positive ce to the same scale that the crank length represents the centrifugal force.

It is evident that, in Fig. 8, $fe = cg$ and $ge = cf$, since they are, respectively, the opposite sides of a rectangle whose diagonal is ce. Then, the horizontal component of ce, or the inertia pressure, is equal to $ce \cos h$, and this remains true for any position of the crank. But, according to the principles of mechanics, the centrifugal force is

$$F_c = \frac{wv^2}{gr}$$

in which F_c = centrifugal force;

w = weight, in pounds, of reciprocating parts, which are generally taken as the piston, piston rod, crosshead, and one-half the connecting-rod;

v = linear velocity of crankpin, in feet per second;

g = 32.16;

r = radius of crankpin circle, in feet.

Then, since ce represents the centrifugal force, the term $\frac{wv^2}{gr}$ may be substituted for ce in the expression $ce \cos h$, and the inertia pressure F is then

$$F = ce \cos h = \frac{wv^2}{gr} \cos h$$

However, $v = 2\pi rn$, in which n is the number of revolutions per second. If N is the number of revolutions per minute, then $n = \frac{N}{60}$ and $v = \frac{2\pi rN}{60}$, and

$$F = \frac{wv^2}{gr}\cos h = \frac{w}{32.16\,r} \times \frac{4\pi^2 r^2 N^2}{3{,}600}\cos h$$
$$= .00034\,wN^2 r\cos h$$

This formula for F gives the inertia pressure for any point of the stroke for an engine with an infinitely long connecting-rod.

EXAMPLE.—An engine with an infinitely long connecting-rod has a cylinder 20″ × 48″; the net forward pressure is 40 pounds at the beginning and 20 pounds at the end of the stroke; the weight of the reciprocating parts is 1,256 pounds. What are the pressures on the crosshead pin at the beginning and end of the stroke, at 50 revolutions per minute?

SOLUTION.—Apply the formula $F = .00034\,wN^2 r\cos h$; in which $w = 1.256$, $N = 50$, $r = 2$, $h = 0°$, and $\cos h = 1$. Therefore,

$F = .00034 \times 1{,}256 \times 2{,}500 \times 2 \times 1 = 2{,}135.2$ lb.

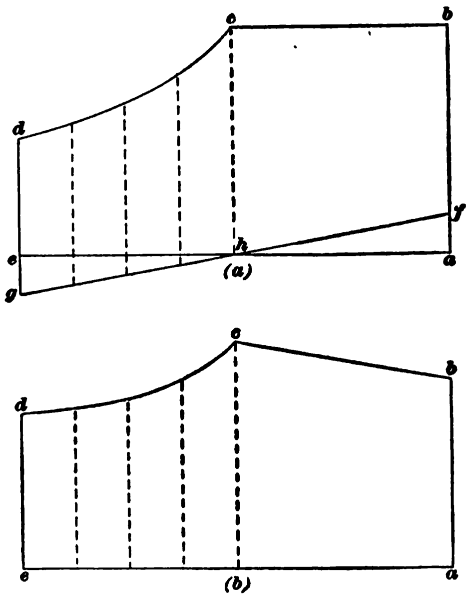


FIG. 9

The pressure on the piston at the beginning of the stroke equals the area, in square inches, multiplied by the pressure, in pounds per square inch, or $.7854 \times (20)^2 \times 40 = 12{,}566.4$ lb. Then the effective pressure urging the crosshead pin forwards at the beginning of the stroke equals $12{,}566.4 - 2{,}135.2 = 10{,}431.2$ lb. Ans.

The pressure on the piston at the end of the stroke, in the above example, equals $.7854 \times (20)^2 \times 20 = 6{,}283.2$ lb., and the effective forward pressure on the crosshead pin, or on the crankpin, at the end of the stroke equals $6{,}283.2 + 2{,}135.2 = 8{,}418.4$ lb. Ans.

The total inertia pressure at the beginning or end of the stroke, if divided by the area of the piston, gives the pressure per square inch, which in this case would be 2,135.2 ÷ 314.16 = 6.8 pounds per square inch.

Let *a b c d e*, Fig. 9 (*a*), represent the diagram of the engine; then, by laying off *a f* and *e g* each equal to 6.8 pounds to the scale of the diagram, and drawing *g f*, which passes through *h* at the point of zero inertia pressure, the figure shows graphically the effect of inertia on the crosshead-pin pressures. The net forward pressures on the crosshead are shown by the lengths of the ordinates between the lines *b c d* and *g h f* to the same scale as that of the work diagrams. These ordinates have been replotted on a horizontal base in Fig. 9 (*b*). Then, any ordinate between *a b* and *d e* represents the actual forward pressure on the crosshead pin for the position of the piston indicated by the point where the ordinate cuts the line *a e*.

9. Effect of the Angularity of Connecting-Rod.—In practice, the inertia pressures differ somewhat from those

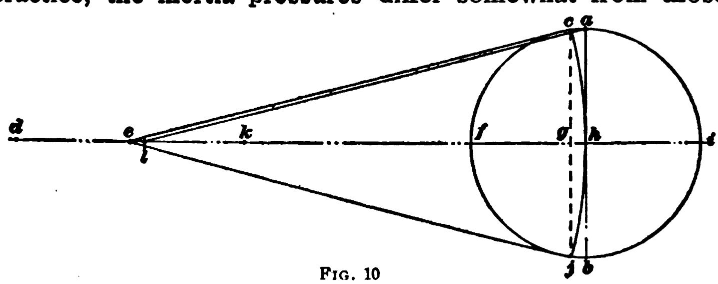


FIG. 10

obtained by means of the formula in Art. **8**, owing to the angularity of the connecting-rod. The effect of this angularity is shown in Fig. 10, in which the circle *a i b f* represents the crankpin circle; *d k*, the travel of the crosshead; and *e*, the mid-position of the crosshead. Owing to the angularity of the connecting-rod, the crankpin stands at *c* or *j* when the crosshead is in mid-position; when the crankpin is at *a* or *b*, the crosshead stands at *l*. Therefore, the crosshead travels a distance *d l*, while the crankpin moves through the 90° arcs *f a* or *b f*, which are the first and fourth quarters of the revolution, and through the distance *l k*, while the crank-pin moves through the arcs *a i* or *i b*, the second and third quarters of the revolution. Since the crank is supposed to move at a uniform speed, it is evident that the piston and other reciprocating parts travel at a higher speed during the first and fourth quarters than during the second and third quarters, and the reciprocating parts must be started more suddenly from rest at the beginning of the forward stroke, and be brought to rest more quickly at the end of the return stroke than at the end of the forward stroke and the beginning of the return stroke. The inertia pressure, therefore, must be greater as the piston starts on its forward stroke and comes to rest on the return stroke than when it comes to rest on the forward stroke or starts on the return stroke. Since the piston changes its direction of motion at the end of the stroke, where its velocity is zero, the inertia pressures are greatest at the ends of the stroke.

The angularity of the connecting-rod has another important effect. Since the velocity of the first part of the forward stroke is greater than the last, the piston, and hence the crosshead also, must reach its maximum velocity before the crank reaches the 90° position. This occurs the instant the crank stands at right angles to the connecting-rod. At this point, the inertia pressure is zero and the inertia curve crosses the base line.

10. In order to find the position of the crosshead for zero inertia pressure, draw a line *a b*, Fig. 11, to represent the center line of the engine, and with *c* as a center construct the crankpin circle *d e b*. Locate the extreme points of the stroke of the crosshead *a* and *f* so that *a d* and *f b* are each equal to the length of the connecting-rod. With *c e* as the crank in any position, draw *g e* at right angles to *c e* and equal in length to the connecting-rod. Then with *c g* as a radius and *c* as a center, draw the arc *g h* locating the point *h* on the center line of the engine. Then, if *h i* be drawn tangent to the circle and *c i* be drawn to the point of tangency, these will represent the connecting-rod and crank when at right angles to each other. It should be noted that the square of the length of the crank plus the square of

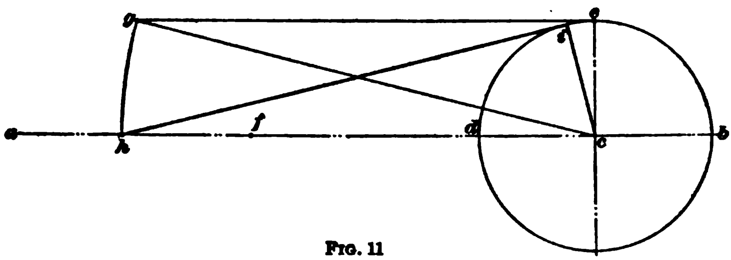


FIG. 11

the length of the connecting-rod is equal to the square of the length of ch, that is, $\overline{ci}^2 + \overline{hi}^2 = \overline{ch}^2$ and $ch = \sqrt{\overline{ci}^2 + \overline{hi}^2}$. As the length of the crank and connecting-rod are always known, it is an easy matter to calculate the position of the center of the crosshead for zero inertia pressure.

11. Effect of the Connecting-Rod on the Inertia Pressure.—In Art. 8, it was determined that in the case of the infinitely long connecting-rod, the inertia pressure is represented by the force $F = .00034\, w N^2 r$, at the beginning of the stroke, since at that position the crank makes no angle with the horizontal.

From the foregoing, it is apparent that the value obtained by this formula must be modified when the connecting-rod is of finite length, as in the practical engine. The best authorities make this correction by the following method, which gives good practical results: When the connecting-rod is n times the length of the crank, the inertia pressure when the crank is nearest to the piston is $\left(1+\frac{1}{n}\right)$ times, and when farthest from the piston is $\left(1-\frac{1}{n}\right)$ times, that of an engine with an infinitely long connecting-rod. That is, with a finite connecting-rod n times as long as the crank, the force, when the crank is nearest the cylinder, is

$$F = .00034\, w N^2 r \left(1+\frac{1}{n}\right) \qquad (1)$$

When the crank is farthest from the cylinder, this force is

$$F = .00034\, w N^2 r \left(1 - \frac{1}{n}\right) \qquad (2)$$

It is the usual practice to locate these three points, the point of zero inertia, and the two end points, on the diagram and to draw an arc of a circle through them for the inertia curve. This is done with very little error and gives very satisfactory results. The method of procedure will be explained by the following example:

EXAMPLE.—A 10″ × 12″ engine runs at 300 revolutions per minute; the weight of the reciprocating parts is 292 pounds and the connecting-rod is 4½ times as long as the crank. Find the inertia pressures at the ends of the stroke and construct the inertia-pressure diagram to some convenient scale.

SOLUTION.—When the crank is nearest the cylinder, according to formula **1**,

$$F = .00034\, w N^2 r \left(1 + \frac{1}{n}\right)$$

$$= .00034 \times 292 \times (300)^2 \times \frac{6}{12}\left(1 + \frac{2}{9}\right) = 5{,}460.4 \text{ lb.}$$

When the crank is farthest from the cylinder, according to formula **2**,

$$F = .00034\, w N^2 r \left(1 - \frac{1}{n}\right) = .00034 \times 292 \times (300)^2 \times \tfrac{6}{12} \times \left(1 - \tfrac{2}{9}\right)$$

$= 3{,}474.8$ lb.

The inertia pressure per square inch of piston area is $P = \frac{F}{A}$, in which A is the area of piston in square inches. Then, for the position of the crank nearest the cylinder,

$$P = \frac{F}{A} = \frac{5{,}460.4}{.7854 \times 10^2}$$

$= 69.5$ lb. per sq. in. Ans.

And for the other extreme position,

$$P = \frac{F}{A} = \frac{3{,}474.8}{.7854 \times 10^2}$$

$= 44.2$ lb. per sq. in. Ans.

To draw the inertia-pressure diagram, lay down $a\,b$, Fig. 12, to represent the length of the stroke to a scale of 3 inches to the foot. Then, choosing a scale of pressures of 50 pounds to the inch, make $a\,c$ equal to $69.5 \div 50 = 1.39$ inch, and $b\,d$ equal to $44.2 \div 50 = .884$ inch. The point e of zero inertia pressure is then located by

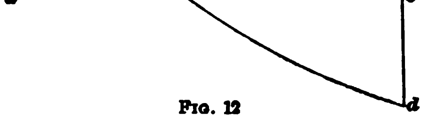


FIG. 12

the method already explained, after which an arc of a circle, ced, is drawn through the three points c, e, and d. The resulting diagram is the required diagram of inertia pressures.

12. Crank-Effort Diagram for High-Speed Engine. Taking the conditions of the example given in Art. **11**, assume further that the initial steam pressure is 100 pounds, gauge; back pressure, 3 pounds, gauge; pressure at the end of compression, 60 pounds, gauge; apparent cut-off, $\frac{1}{3}$ stroke; clearance, 10 per cent. Also, assume that the curves of expansion and compression are equilateral hyperbolas. From these data, the tangential-effort diagram may be accurately constructed, taking into account the net steam pressures, the inertia pressures, and the effect of the angularity of the connecting-rod.

It is necessary, first of all, to construct the indicator diagrams as in Fig. 13 (a). Choose a scale of pressures of say 50 pounds to the inch and a scale of horizontal distances of 3 inches to the foot. In order to bring the illustrations within the size of the page, Figs. 12, 13, and 14 are drawn to two-thirds size, instead of to the scales stated. The scale dimensions may be obtained by multiplying the dimensions in these figures by $\frac{3}{2}$. Draw any horizontal line xy as an atmospheric line, and on it lay off ab equal to 3 inches, which will then represent the stroke of the engine, or 1 foot. At a and b erect perpendiculars to xy. Since the boiler pressure is 100 pounds, and the scale of pressures is 50 pounds to the inch, lay off bc equal to $100 \div 50 = 2$ inches. Then, from the point c draw a line parallel to xy until it intersects the perpendicular through a at d.

Next, make $ce = \frac{1}{3}\,cd$, since the apparent cut-off is $\frac{1}{3}$, and e will represent the point of cut-off. The clearance being 10 per cent., lay off bg equal to 10 per cent. of ab, and erect the perpendicular gf, which will be the clearance line. Below ab, at a distance representing the pressure of the atmosphere, or $14.7 \div 50 = .294$ inch, draw the vacuum line, cutting fg, produced, at o. Then o is the point of zero pressure and zero volume. On bc, lay off bh to represent 60 pounds; that is, $bh = 60 \div 50 = 1.2$ inches. The back pressure is 3 pounds, gauge, and so the back-pressure line mn is drawn parallel to ab and at a distance of $\frac{3}{50}$ inch, say $\frac{1}{16}$ inch, above it. Then, from the point e, draw an equilateral

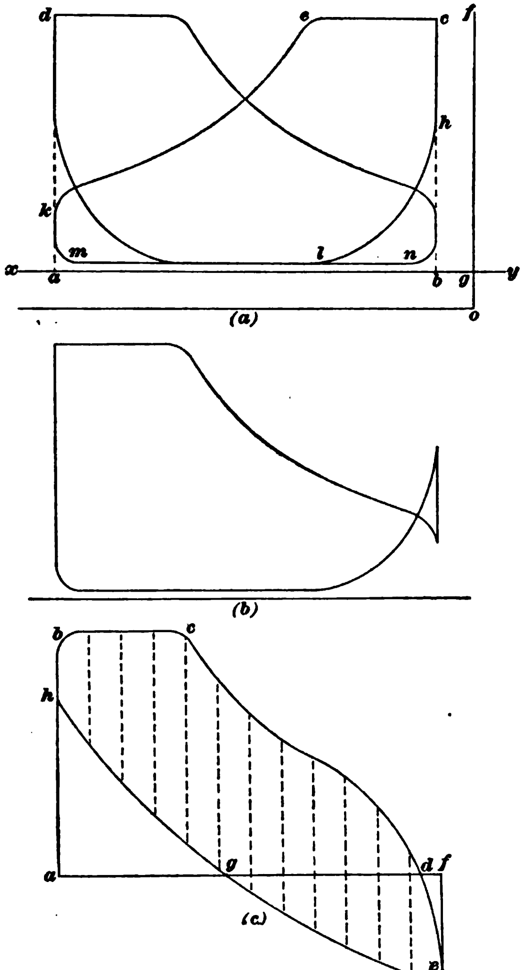


FIG. 13

hyperbola. This will be the expansion line, and it should be rounded off slightly at cut-off and at release. Also, at the point where the release line joins the back-pressure line, it should be slightly rounded, as shown. From *h*, another equilateral hyperbola is drawn to represent the compression curve. Then the theoretical diagram of the engine is *c e k m l h*. By the same method, on *a b* as a base, construct a similar diagram for the other end of the cylinder. Fig. 13 (*b*) shows the second step, namely, the combining of the expansion line of one diagram with the back-pressure line of the other to give the diagram of net steam pressures on the piston. This is accomplished by the same means through which Fig. 6 was derived from Fig. 5.

The next step is to lay out the diagram in Fig. 13 (*b*) to a straight base, just as Fig. 7 was obtained from Fig. 6. When this is done, the diagram *a b c d e f a*, Fig. 13 (*c*), is obtained, the ordinates of which represent the net forward steam pressures per square inch of piston area.

Now, it becomes necessary to take into account the inertia pressures as modified by the angularity of the connecting-rod. The inertia-pressure diagram has already been determined in connection with the example in Art. **11.** Hence, on Fig. 13 (*c*), lay off *a h* and *f i* equal, respectively, to *a c* and *b d* of Fig. 12. Also, make *a g*, Fig. 13 (*c*), equal to *a e*, Fig. 12, and draw *h g i* an arc of a circle. Then the lengths of the ordinates between the curves *b c d e* and *h g i*, measured vertically, represent the actual net pressures per square inch on the crosshead pin for all positions of the piston on its forward stroke.

13. Next, draw a horizontal line *1–13*, as in Fig. 14 (*a*), equal in length to *a f*, Fig. 13 (*c*), and on it construct the diagram of the net pressures on the crosshead pin to a straight base, which is represented by the outline *1 b d h 13*. This is done by measuring the ordinates between the lines *b c d e* and *h g i*, Fig. 13 (*c*), and laying them off in the same order vertically above *1–13*, Fig. 14 (*a*). The diagram *1 b d h 13* is then the one from which the crank-effort diagram is to be constructed, by the graphic method explained in Art. **4.**

The line *1–13*, Fig. 14 (*a*), is extended to the right indefinitely, and *1–1′* is laid off equal to $4\frac{1}{2}$ times the crank length.

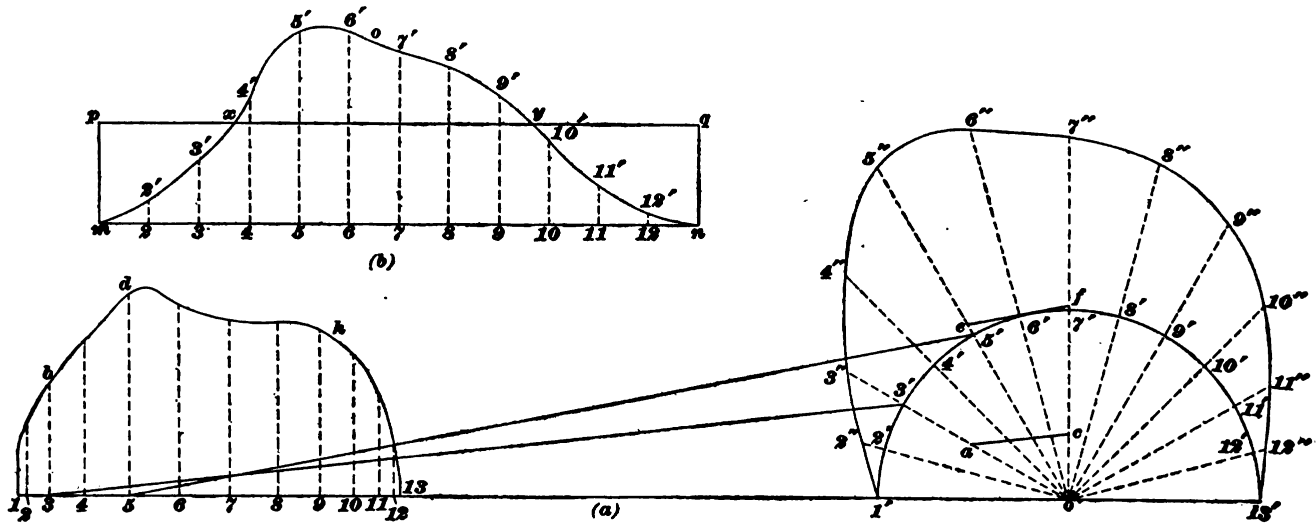


Fig. 14

The crank length is evidently one-half the distance *1–13*. Hence, *1–1′* is equal to 2¼ times the distance *1–13*. Next, make *1′–13′* equal to *1–13*, and with the middle point *o* as a center, describe the semicircle *1′–7′–13′*, which is the path of the crankpin. Then, divide the semicircle into a number of equal parts, say twelve, marking the points of division *1′*, *2′*, *3′*, etc. Now, with a radius equal to the length *1′–1* of the connecting-rod, and with centers at *1′*, *2′*, *3′*, etc., in succession, strike arcs cutting the line *1–13* at the points *1*, *2*, *3*, etc. Then the points *1*, *2*, *3*, etc. represent the positions of the crosshead corresponding to the crank-positions *o 1′*, *o 2′*, *o 3′*, etc., and from these crosshead positions, *1*, *2*, *3*, etc., vertical lines are drawn. The lengths of the vertical lines included between the line *1–13* and the curve *b d h* represent the actual pressures per square inch of piston area, acting on the crosshead pin at the corresponding positions of the crosshead.

To find the tangential pressure at the crankpin, take, for example, the position *3* of the crosshead, with the crank at *o 3′*. Draw the connecting-rod *3′–3*. Now, on the radius *o 3′*, lay off *o a* equal to the ordinate *3 b* of the diagram of crosshead-pin pressures, and from *a* draw a line parallel to the connecting-rod *3′–3*, until it cuts the vertical line *o 7′* at *c*. Then the length *o c* represents the tangential pressure on the crankpin, when the crosshead is at *3*, to the same scale that *3 b* represents the pressure on the crosshead pin.

Now, produce the center line of the crank *o 3′* outwards beyond the semicircle, and make *3′–3″* equal in length to the intercept *o c*. Then *3″* is one point on the tangential effort curve.

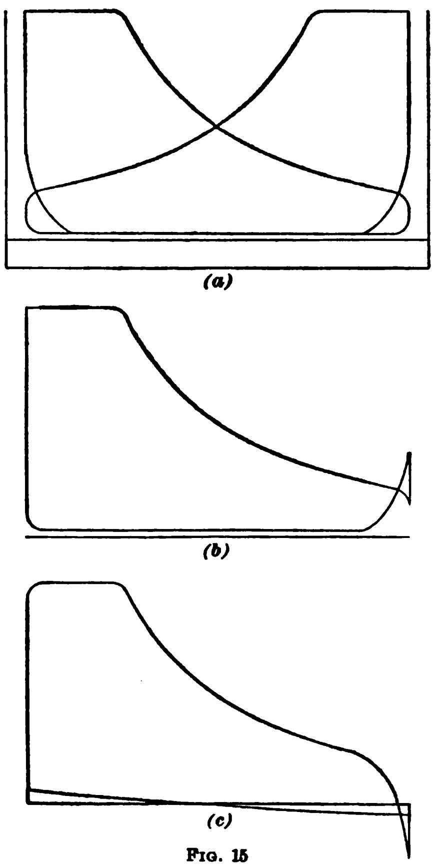
Similarly, choosing the position *5* of the crosshead, the ordinate *5 d* should be laid off on *o 5′* extended, locating the point *e*. Then, from *e*, a line should be drawn parallel to the connecting-rod position *5′–5* until it cuts *o 7′* at *f*. Then *o f* will represent the tangential effort per square inch of piston for position *5* of the crosshead. On *o 5′* produced, lay off *5′–5″* equal to the length of the intercept *o f*, and *5″* will be another point on the crank-effort curve.

By the same process, a series of points may be located, and a curve drawn through the points *1′–2″–3″–4″* . . . *13′* will be the crank-effort curve. Any intercept between the semicircle *1′–7′–13′* and this curve, measured on a radial line through *o*, will represent the tangential pressure per square inch of piston at that crank position.

For convenience in making calculations, the crank-effort curve in Fig. 14 (*a*) should be laid out to a straight base. To do this, draw a horizontal straight line, and on it lay off *m n*, Fig. 14 (*b*), equal in length to the semicircle *1′–7′–13′*; that is, equal to π times the radius *o 1′*, Fig. 14 (*a*). Then divide *m n* into the same number of equal parts as there are in the semicircle, namely, twelve. At the points of division erect perpendiculars to *m n*, and on these perpendiculars lay off, in succession, the ordinates of the crank-effort curve in (*a*); that is, make *2–2′*, *3–3′*, *4–4′* etc. of (*b*) equal, respectively, to *2′–2″*, *3′–3″*, *4′–4″*, etc. of (*a*), and through the points *2′*, *3′*, *4′*, etc., thus located, trace a smooth curve. The resulting diagram *m o n* is the crank-effort diagram, or diagram of tangential pressures to a straight base.

The next step is to find the mean ordinate of *m o n*. This is done by means of a planimeter, or by erecting ordinates at the middle points of the equal divisions, adding their lengths, and dividing the sum by the number of ordinates measured. This mean ordinate *m p* is then laid off vertically and the line *p q* drawn parallel to *m n*. The height *m p* represents the average resistance of the load on the engine, on the assumption that the resistance is uniform. Then, the area *m p q n* is equal to the area *m o n*, or the average effort of the engine per stroke is equal to the resistance of the load in the same time. This is as it should be, since, if the effort of the engine were greater than the load resistance, the engine would run faster and faster; and if it were less than the load resistance, the engine would gradually slow down and stop.

14. Crank-Effort Diagram for Corliss Engine. Doubtless the crank-effort diagram of a slow-speed engine will show some points of difference from that of a high-speed engine. For example, let it be required to work out the crank-effort diagram for a 24″ × 36″ Corliss engine running at 80 revolutions per minute; steam pressure, 120 pounds, gauge; apparent cut-off, $\frac{1}{4}$ stroke; clearance, 5 per cent.; back pressure, 3 pounds, gauge; pressure at the end of compression, 45 pounds, gauge. Assume the curves of expansion and compression to be equilateral hyperbolas. The weight of the reciprocating parts is 834 pounds and the connecting-rod is 8 feet long.



(a) (b) (c)

FIG. 15

By the use of the formulas in Art. **11**, the inertia pressures are found to be 3,233 and 2,212 pounds, respectively, corresponding to 7.14 and 4.89 pounds per square inch of piston area. Then, the several successive diagrams, Fig. 15 (*a*), (*b*), and (*c*), and Fig. 16 (*a*) and (*b*), are drawn, pursuing the same methods as explained in the previous article. It will be noted, however, that the inertia pressure at the end of the stroke is considerably less than the net steam pressure. As a consequence, the diagram of net pressures on the crosshead pin, Fig. 16 (*a*), extends below the line of zero pressure, in which it differs from the diagram of net pressures in Fig. 14 (*a*). And because

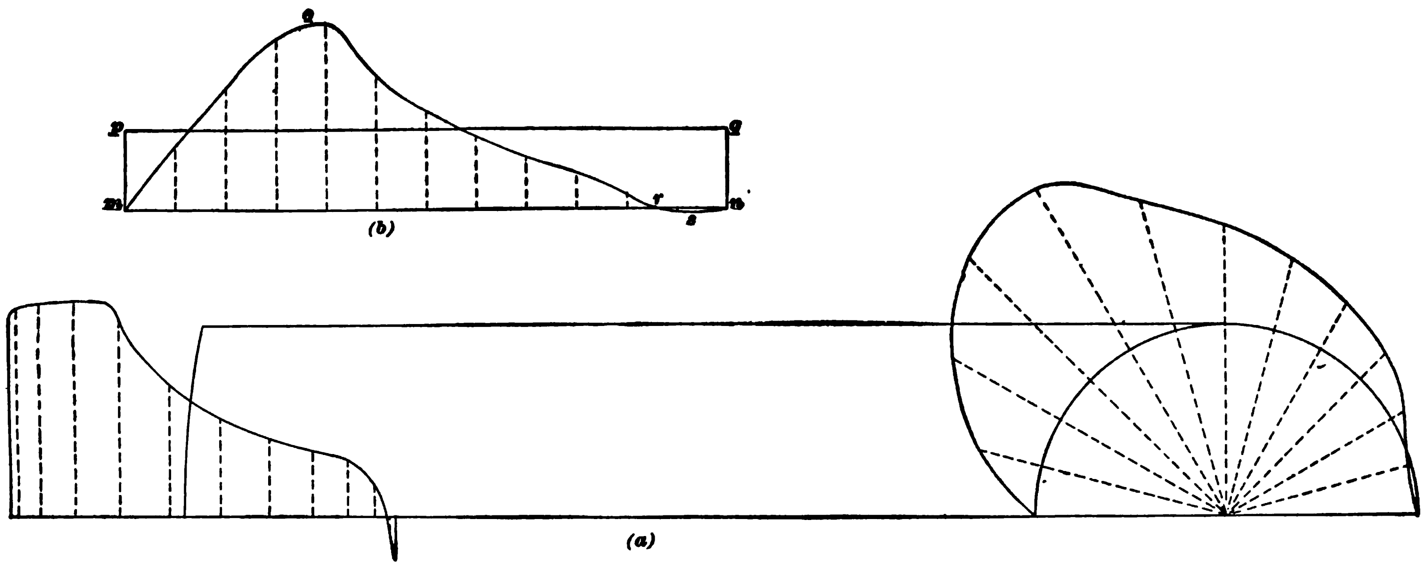


FIG. 16

of this, the tangential-effort curves, Fig. 16 (*a*) and (*b*), also extend below their base lines. In finding the net mean ordinate of Fig. 16 (*b*), therefore, it is necessary to find the difference of the areas *mor* and *rsn*, and divide it by the total length *mn*; the result will be the mean height *mp*. The reason for this is apparent, since all areas below the base line are negative; that is, they represent work done on the steam instead of work done by the steam.

15. Tangential Pressures and Initial Stresses in Multiple-Expansion Engines.—The load on the engine, or the resistance against which the engine works, is usually constant. That is, it requires a practically constant force to drive the shafting or machines. It is, therefore, very desirable that the force turning the crank-shaft should be as nearly constant as possible. It has been shown by the diagrams of

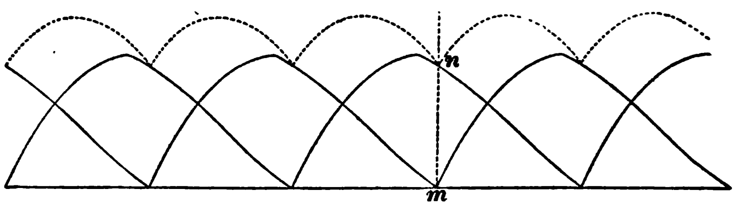


FIG. 17

Figs. 14 and 16 that this tangential force is far from being constant; that at the dead points it is zero and that near the middle of the stroke it is greatest. It can now be shown why compound and duplex engines, which have their cranks at right angles, possess advantages over the simple engine. In Fig. 17 is shown a crank-effort diagram for a cross-compound engine with cranks at right angles. When one crank is beginning its stroke, the other is at the middle of the stroke, and vice versa. Therefore, when the curve of one crank touches the base line, as at *m*, the other will be at or near its highest position, as at *n*. Now, adding the corresponding ordinates of the two curves together, the dotted curve is obtained, which, therefore, represents the total tangential pressure tending to turn the crank-shaft. It is apparent at a glance how much more nearly constant is the tangential pressure of the compound-engine diagram than the tangential pressure of the simple-engine diagram. Consequently, with the same steam pressure and weight of fly-wheel, the compound or duplex engine with cranks at 90° will run more steadily than the simple engine, while a triple-expansion engine with three cranks, making angles of 120° with one another, will run more steadily than the compound.

16. A tandem compound engine, also, has a mechanical advantage over the simple engine, as a little consideration will show. A compound engine is equivalent to a simple engine whose cylinder is of the same dimensions as the low-pressure cylinder, and which expands its steam the same number of times as does the compound.

For convenience of illustration, assume the cut-off in the high-pressure cylinder to be $\frac{1}{2}$, and that the ratio of the volumes of the two cylinders is 1 : 4; that there is no clearance and no receiver; that there is no loss of pressure due to wiredrawing, friction, etc., and, finally, that the steam is carried full stroke in the low-pressure cylinder. The total number of expansions is eight. Assume the steam pressure to be 120 pounds, absolute, and the absolute back pressure to be 15 pounds. Denote the area of the large piston by A; then, the area of the small piston is $\frac{A}{4}$. The terminal pressure in the small cylinder is, evidently, 60 pounds, and this is the initial forward pressure in the large cylinder and the initial back pressure in the small cylinder.

Consider the simple engine first. The initial forward pressure is 120 pounds and the back pressure 15 pounds; hence, the net pressure urging the piston ahead is $120 - 15 = 105$ pounds. The total force acting on the piston is $105A$.

Considering the compound engine, the initial forward pressure in the small cylinder is 120 pounds; the back pressure 60 pounds, and the net pressure $120 - 60 = 60$ pounds. Since the area of the small piston is $\frac{A}{4}$, the total force tending to drive the small piston forwards is $60 \times \frac{A}{4} = 15A$. The initial forward pressure in the low-pressure cylinder is 60 pounds, the back pressure is 15 pounds, and the net pressure urging the piston ahead is $60 - 15 = 45$ pounds. The total force tending to drive the large piston forwards is $45A$. The total initial force acting on both pistons of the compound is $45A + 15A = 60A$; in the simple engine, it was $105A$. Since the greatest stresses occur when the forces that produce them are greatest, it is evident, from the foregoing, that the various parts of a compound engine (connecting-rod, crank, shaft, etc.) will not need to be made so large as in a simple engine having the same mean effective pressure, and the volume of whose cylinder equals the volume of the low-pressure cylinder. This, however, is a disadvantage when the engine has a very heavy load to start, as in the case of a locomotive, for in that case the compound engine might not be able to start, although it could keep itself in motion after it had once been started.

FLYWHEELS

17. The office of the **flywheel** is somewhat similar to that of the governor, since each is used to obtain regularity of speed. It is the duty of the governor to adjust the effort of the engine to any large or permanent variation of the load, such as would be caused by throwing the machinery in or out of gear. It is the duty of the flywheel, on the other hand, to adjust the effort of the engine to sudden fluctuations of the resistance, which may occur during a single stroke of the engine. It is also the duty of the flywheel to equalize the varying tangential effort on the crankpin by storing energy while the piston is in the middle of the stroke, where the crank-effort is greater than the resistance, and restoring it when the crank is at the dead point, where the tangential effort is zero.

In the tangential-effort diagram, Fig. 18, the line AB represents the travel of the crankpin during half a revolution. While the crankpin is moving from A to S the effort of the engine is increasing, yet it is always less than the average resistance AM. Since a smaller force cannot overcome a larger one, it is evident that the engine must be receiving energy from some other source than the steam pressure while the crankpin is traveling from A to S. During this period, a part of the energy stored in the moving parts is being restored, so as to enable the effort at the crankpin to equal the resistance.

At S, the effort equals the resistance; and from S to T, it is greater, as shown by the rising of the curve above the line MN. Hence, as the engine runs at a practically uniform speed, it is evident that the excess energy represented by the area

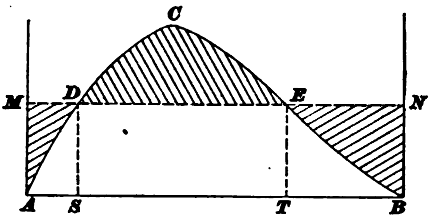


Fig. 18

CDE is stored in the moving parts, to be given back during those periods in the stroke when the effort is less than the resistance. From T to B, the effort is again less than AM and the moving parts supply the extra energy necessary to maintain uniform speed. Since the energy absorbed by these parts during a part of the stroke is equal to that given back again during the remainder of the stroke, it follows that area CDE = area AMD + area BNE.

The kinetic energy of a moving body is $\frac{Wv^2}{2g}$. In order to give up energy, the v of this expression must be diminished. That is what actually takes place. All the moving parts of the engine (the reciprocating parts, shaft, and flywheel) slow down a little, and in so doing give up enough of their kinetic energy to overcome the resistance and carry the engine past the dead center. From S to T, the effort is greater than the resistance, and, consequently, the surplus energy represented by the area CDE is stored up in the moving parts—that is, their velocity is increased during that part of the stroke, and with it their kinetic energy. For the remainder of the stroke, the moving parts again slow down and give up an amount of energy represented by the area BNE.

In practical engine design, it is assumed that all this energy is stored, and restored, by the rim of the flywheel, the effect of the other moving parts being so small in comparison that they may be neglected. Any effect they may have simply adds so much more to the steadiness of the engine.

18. The **weight of the flywheel rim** may be found in the following manner:

Let V_1 = greatest velocity of crankpin, in feet per second;
V_2 = least velocity of crankpin, in feet per second;
V_0 = average velocity of crankpin, in feet per second;
W = required weight of flywheel rim, in pounds;
H = number of foot-pounds per square inch of piston represented by area CDE;
A = area of piston, in square inches;
n = ratio between the mean radius of flywheel rim and the length of crank.

$$E = \frac{V_1 - V_2}{V_0} = \text{coefficient of unsteadiness} \qquad (1)$$

A, of course, is known, and n depends on the assumed diameter of the flywheel.

The average velocity V_0 is known, since it is $\frac{\pi}{2}$ times the piston speed, in feet per second. V_1 and V_2 are assumed, so that the fraction $\frac{V_1 - V_2}{V_0} = E$ shall not exceed a certain value.

The following values of E agree well with those ordinarily accepted in practice:

Pumping engines	$\frac{1}{20}$
Engines driving machine tools	$\frac{1}{35}$
Engines driving textile machinery	$\frac{1}{40}$
Engines driving spinning machinery	$\frac{1}{50}$ to $\frac{1}{100}$
Engines driving electric machinery	$\frac{1}{125}$ to $\frac{1}{150}$

H is found directly from the diagram by multiplying the area CDE, in square inches, by the vertical scale of pressures and the horizontal scale of distances.

Let a = area of CDE, in square inches;

h = scale of pressures used in constructing crank-effort diagram, which is the same as the scale of the indicator spring used on the indicator diagram;

h' = scale of horizontal distances, or the number of feet of crankpin travel represented by each inch of length of AB; that is, it is equal to the crankpin travel in feet, during half a revolution, divided by the length AB in inches

Then, $$H = a\,h\,h' \qquad (2)$$

At the point D, the crankpin has the velocity V_2; since the mean radius of the flywheel rim is n times the length of crank, the velocity of the rim at D must be nV_2. Likewise, at E, the crankpin will have its greatest velocity V_1, and the flywheel rim the velocity nV_1.

The kinetic energy of the flywheel rim at D is $\frac{W(nV_2)^2}{2g}$; and at E, it is $\frac{W(nV_1)^2}{2g}$. The kinetic energy stored up in passing from D to E is, therefore, $\frac{Wn^2V_1^2}{2g} - \frac{Wn^2V_2^2}{2g}$ $= \frac{Wn^2}{g}\left(\frac{V_1^2 - V_2^2}{2}\right)$ foot-pounds. But this kinetic energy stored up is equal to the work represented by the area CDE. Hence,

$$\frac{Wn^2}{g}\left(\frac{V_1^2 - V_2^2}{2}\right) = A \times H \qquad (1)$$

But $\frac{V_1 + V_2}{2} = V_0$, the average velocity, and $\frac{V_1 - V_2}{V_0} = E$; therefore,

$$\left(\frac{V_1 - V_2}{V_0}\right)\left(\frac{V_1 + V_2}{2}\right) = \frac{V_1^2 - V_2^2}{2V_0} = EV_0$$

or $$\frac{V_1^2 - V_2^2}{2} = EV_0^2$$

Substituting this in (1), $\frac{Wn^2}{g}\left(\frac{V_1^2 - V_2^2}{2}\right) = \frac{Wn^2EV_0^2}{g} = AH$;

whence, $$W = \frac{AHg}{n^2EV_0^2} \qquad (3)$$

EXAMPLE.—Given a single-cylinder engine making 100 revolutions per minute, with a stroke of 3 feet and a cylinder diameter of 20 inches. A diagram similar to that of Fig. 18 is drawn, and it is found that the area of the portion CDE is .86 square inch. The diagram was drawn so that a vertical height of 1 inch represents 40 pounds pressure per square inch, and a horizontal distance of 1 inch represents a crank-pin travel of $1\frac{1}{2}$ feet. Assume the coefficient of unsteadiness E to be $\frac{1}{40}$ and the ratio n as 4. Find the necessary weight of the flywheel rim.

SOLUTION.—Piston speed is $\frac{2 \times 3 \times 100}{60} = 10$ ft. per sec. Average speed of crankpin is speed of piston $\times \frac{\pi}{2} = 10 \times \frac{3.1416}{2} = 15.7$ ft. per sec. $= V_0$. Area of piston is $.7854 \times 20^2 = 314.16$ sq. in. Work per square inch of piston represented by area $CDE = .86 \times 40 \times 1\frac{1}{2} = 51.6$ ft.-lb. $= H$. By formula **3**,

$$W = \frac{AHg}{n^2 E V_0^2} = \frac{314.16 \times 51.6 \times 32.16}{4^2 \times \frac{1}{40} \times 15.7^2} = 5{,}288 \text{ lb.} = 2.644 \text{ tons. Ans.}$$

DYNAMOMETERS

19. Dynamometers are devices for measuring power. They are divided into two main classes—*absorption dynamometers* and *transmission dynamometers.*

The most common form of **absorption dynamometer** is the *Prony brake*, which consists simply of a friction brake designed to absorb in friction, and measure, the work done by a motor or the power given out by a shaft.

A **transmission dynamometer** is used to measure the power required to drive a machine or do other work. To determine the power required to run the shafting in a mill, a transmission dynamometer would be interposed between the shafting and the source of power, and by suitable belt connections the shafting would be driven through the dynamometer, from which the power may be determined.

PRONY BRAKE

20. Fig. 19 represents a simple and common form of **Prony brake.** It consists of two wooden blocks A and B that are clamped together on a pulley P, by the bolts and thumb nuts c, c. The same bolts clamp an arm L to the upper block, from which a scale pan, bearing a known weight W, is suspended. The distance R from the center of the pulley to the perpendicular through the point from which the scale pan is suspended is also known. The counterweight w should be so adjusted as to just balance the extra length of L on the right and the weight of the scale pan.

Suppose the pulley to revolve left-handed and the nuts c, c to be tightened until, with a weight W in the scale pan, the lever L will remain stationary in a horizontal position. *The foot-pounds of work absorbed in a given time by the brake can then be found by multiplying the weight W by the circumference of a circle whose radius is R feet and by the number of revolutions of the pulley in that time.*

It is important to note that neither the diameter of the pulley nor the pressure with which the blocks clamp the

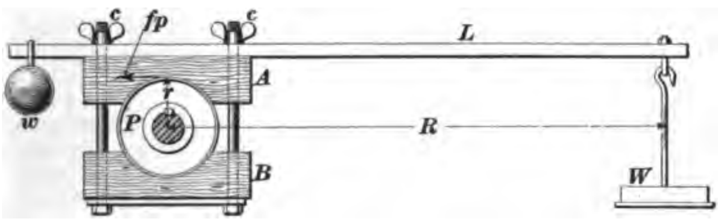


FIG. 19

pulley enter into the calculations at all. For, letting p represent the pressure and f the coefficient of friction between the blocks and the pulley, the force at the face of the pulley tending to resist its rotation will be fp. The force tending to keep the lever L from turning, however, is W, and as the bolts are adjusted so that L remains constantly in a horizontal position, the moments of these two forces about the center of the pulley are equal, or $fpr = WR$.

Now, the work done at the face of the pulley is equal to the force exerted times the number of feet passed through, or, calling N the number of revolutions per minute, it is $2\pi r \times fp \times N$. This, it will be seen, is the first member of the above equation multiplied by $2\pi N$. Multiplying the second member by $2\pi N$, also, to keep both members equal, the result obtained is another expression for the work absorbed, $2\pi R \times W \times N$; this is the formula used in calculations. Hence, let

B. H. P. = number of horsepower absorbed, or the brake horsepower;
R = length, in feet, of lever arm about center of shaft;
W = weight in scale pan;
N = number of revolutions per minute.

Then, $$\text{B. H. P} = \frac{2 \times 3.1416 \times R \times W \times N}{33{,}000}$$

This may be reduced to B. H. P. = .0001904 $R W N$.

EXAMPLE.—A brake with an arm R 6 feet long was placed on the flywheel of an engine. If the engine ran at 200 revolutions per minute, what power did it develop when the brake balanced with 14 pounds in the scale pan?

SOLUTION.—Applying the formula,
B. H. P. = .0001904 × 6 × 14 × 200 = 3.199 H. P. Ans.

21. Brakes are often constructed of a metal band that

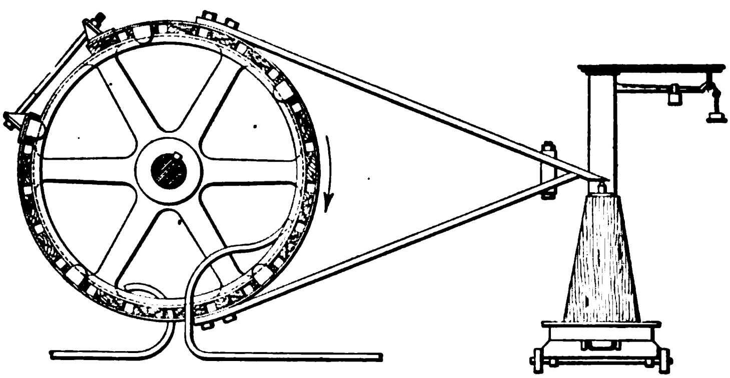


FIG. 20

extends entirely around the pulley, the rubbing surface being formed of blocks of wood fitted to the inside of the band. A weight arm is attached to one side of the pulley and the friction is varied by means of a bolt and nut used to connect the two ends of the band.

Instead of hanging weights in the scale pan, the friction may be weighed on a platform scale, as shown in Fig. 20.

In order to use this brake, the pressure of the arm on the scale should be taken when the machine is at rest; this pressure should then be deducted from the pressure on the scale when the machine is running, and the difference thus obtained used as the value of W in the formula of Art. **20.**

It is essential that these brakes should be well lubricated, and for all except small powers means must be provided for

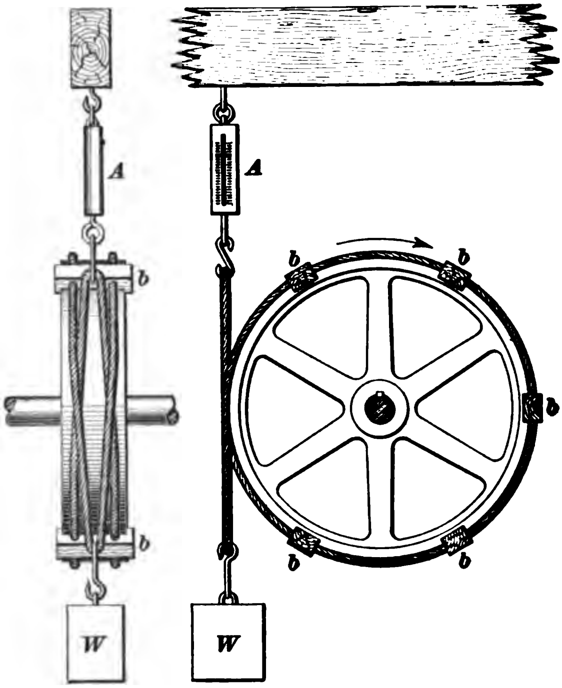


FIG. 21

conducting away the heat generated by friction. If there are internal flanges on the brake wheel, water can be run on to the inside of the rim, the flanges serving to retain the water at the sides and centrifugal force to keep it in contact with the rim. A funnel-shaped scoop can be used to remove the water. It should be attached to a pipe and placed so as to scoop out the water, which should flow continuously. This arrangement is shown in Fig. 20.

22. A **rope brake,** like that in Fig. 21, will give good results. The figure shows the construction so clearly that no description is necessary. To obtain the brake load, subtract the brake pull, as registered by the spring balance, from the weight. In this case, the lever arm is r, equal to the radius of the pulley + ½ the diameter of the rope. If this radius be given in inches, the formula of Art. **20** becomes

$$\text{B. H. P.} = .00001586\, r\, W N$$

TRANSMISSION DYNAMOMETERS

23. There are several forms of transmission dynamometers regularly manufactured, but as rules for their use

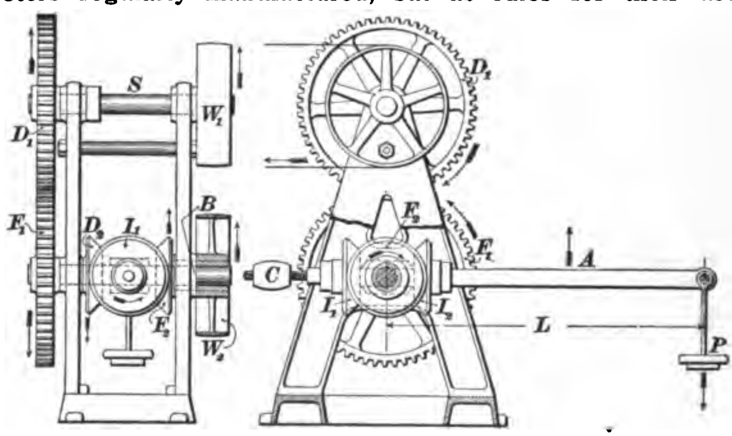


FIG. 22

always accompany the machines, only one form, the **differential dynamometer,** will be described here. The side and end elevations of this dynamometer are shown in Fig. 22. W_1 is a pulley to be belted up with the pulley on the line shaft, or other source of power, that drives the machine to be tested. W_2, placed for convenience in the same plane with W_1, is of the same size as W_1 and is to be belted to the driving pulley on the machine. Connection between W_1 and W_2 is made by means of the shaft S, running in bearings on the frame, the two gears D_1, F_1, of equal diameters, and the differential gearing shown.

Of this latter, D_2 is keyed to the same shaft as F_1. F_2 is loose on the shaft and has a long hub reaching through the bearing B and through the hub of the pulley W_2 to which it is fastened. D_2 and F_2 are connected by the two miter gears I_1 and I_2 in the usual way, having their bearings on the arm A, and being free to turn about the lower shaft. There is a scale pan at the right-hand end of the arm hung from a knife edge, and it is clear that if a weight P be put in the pan sufficient to hold it down, I_1 and I_2 will act simply as idlers and F_2 will turn in the opposite direction and with the same speed as D_2. Hence, W_1 will turn in the same direction as W_2, and with the same speed.

A counterweight C is provided for balancing the arm when the dynamometer is at rest and there are no weights in the scale pan. To find the amount of power transmitted, the length L of the arm, the weight P in the scale pan, and the number of revolutions must be known. The arm is generally of such a length that the circumference that the knife edge would describe if the arm revolved about the shaft would be some even number of feet, say ten.

If pulley W_1 is turned and F_2 is at the same time held fast so that it cannot turn, the arm will make one-half the number of revolutions made by W_1. Twice as great a weight, therefore, will be required in the scale pan to keep the arm stationary as would be necessary if the arm made the same number of revolutions. Hence, applying the principle of the Prony brake, and supposing that the circumference of the circle whose radius is L is 10 feet, *2 pounds in the scale pan will correspond to $1 \times 10 = 10$ foot-pounds of power transmitted per revolution of the shaft.*

EXAMPLES FOR PRACTICE

1. The lever arm of a Prony brake is 4.5 feet; weight in scale pan, 2 pounds 4¼ ounces; revolutions per minute of pulley, 160. Required the brake horsepower. Ans. .313 H. P.

2. A rope brake is used on a pulley 36 inches in diameter. The diameter of the rope is ⅝ inch, revolutions of pulley per minute, 200. How many horsepower are absorbed, the weight being 210 pounds and the balance reading being 5 pounds? Ans. 11.9 H. P.

3. An actual test with a rope brake showed a mean brake horsepower of 15.23; the mean number of revolutions of the wheel per minute was 205, the weight used was 157 pounds, and the mean back pull on the balance was 4 pounds. What was the length of the lever arm? Ans. 30.6 in.

STEAM-ENGINE GOVERNORS

MECHANICS OF THE GOVERNOR

TYPES OF GOVERNORS

1. Speed Regulation.—In order that the speed of an engine may be well regulated, a continuous governing action must take place. This is necessary in order that the work done by the steam in the cylinder shall be kept equal to the load on the engine.

All **steam-engine governors** control the speed of the engine by regulating the amount of steam admitted to the cylinder. This may be done in either of two ways: (1) The governor may vary the pressure of the steam by means of a throttling valve, leaving the point of cut-off the same; (2) the governor may vary the volume of steam admitted to the cylinder by changing the point of cut-off, the initial pressure remaining constant.

In both cases, the action of the governor depends on a change of speed of the engine. The governor is driven from the main shaft, either directly or by belt or other suitable connection. A change of speed of the engine thus causes at once a corresponding change of speed of the governor. The forces set up by this change cause a change in the relative positions of the parts of the governor, and the motion of these parts, properly transmitted, produces the necessary regulation.

2. General Classification.—For the purposes of this treatise, governors will be considered as being of three classes: (1) Pendulum or flyball governors; (2) centrifugal governors; (3) inertia governors. In order to explain the essential features and the distinctive differences of the three types, a simple form of each will be described.

3. A common form of **flyball governor** is illustrated in Fig. 1. The vertical spindle *a* is driven, by belt and bevel gears, from the engine shaft, and has a speed of rotation at all times proportional to that of the main shaft. At some definite speed of the engine, the balls *b, b* revolve around the spindle in the horizontal plane *XX*. If the speed increases, the greater centrifugal force developed will cause the balls to swing outwards and upwards, and then they will move in a higher plane *YY*. This upward movement of the balls is transmitted through the arms *c, c* to the sleeve *d*, which rises on the spindle. The vertical motion of the sleeve is then employed to partly close the throttling valve or else so adjust the valve gear as to cause an earlier cut-off. With a decrease in the speed of the engine, the reverse occurs.

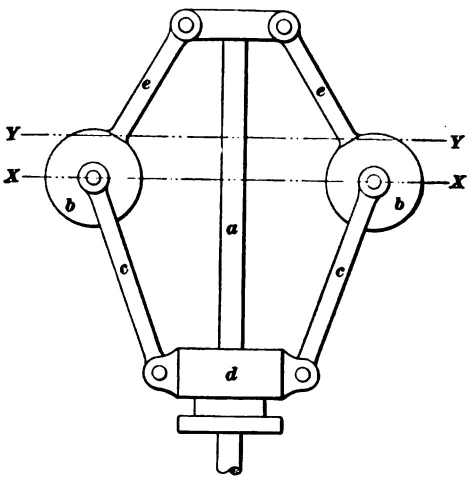


FIG. 1

4. The **centrifugal governor** illustrated in Fig. 2 is a form of shaft governor. A weight *a* is fastened to one arm of a bell-crank lever that is pivoted at *b* to an arm of the flywheel. A spring *c* tends to draw the weight *a* toward the center of the wheel. At a given speed, the center of the weight *a* will be at a certain distance from the center *d* of the shaft, the outward pull of the weight, due to centrifugal force, being counterbalanced by the inward pull of the spring. If the speed decreases, the centrifugal force becomes less, and the tension in the spring draws the weight inwards until a new balance is effected. This motion causes the bell-crank to turn on the pivot *b*, and the resulting motion of the free arm *e* is transmitted to the valve gear so as to give a later cut-off.

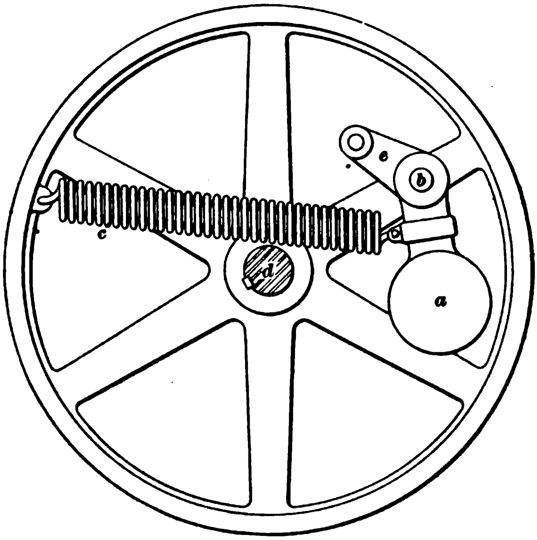


FIG. 2

5. Fig. 3 shows the essential features of the **inertia governor.** A weight arm, consisting of a bar *a* carrying two weights *b*, *b* is pivoted at *c* to an arm of the flywheel. The point *c* in this case is also the center of gravity of the weight arm, and this point always remains at the distance *dc* from the shaft center *d*. If the wheel turns at a uniform speed in the direction of the arrow, there will be no change of position of the weight arm relative to the wheel, and all parts will turn about *d* at the same rotative speed. If a sudden increase of load should cause the engine to slow down, the weight arm, by reason of its inertia, will turn about the pivot *c* in the direction of the arrow; for the weights *b*, *b*, having the same rotative speed as the wheel before the change of load occurred, will tend to keep on revolving at that speed, which will result in swinging the weight arm forwards on the pivot *c*. This motion of the weight arm relative to the wheel is utilized to alter the point of cut-off. Should the speed increase, the weight arm will

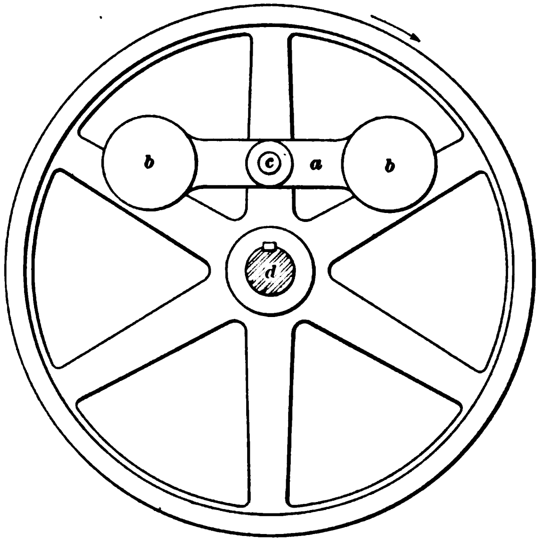


FIG. 3

lag behind and will turn in the opposite direction about the point *c*.

PENDULUM OR FLYBALL GOVERNORS

6. The Simple Revolving Pendulum.—If a ball *a*, Fig. 4, is attached to a cord *b* whose upper end is fastened at *c* to a vertical spindle *c o*, and if the ball is then swung in a circular path about the spindle, the string *b* will describe the surface of a cone, the apex of which is at *c*, and the base of which is the circle described by the center of the ball. This device is known as the **conical pendulum or simple revolving pendulum.** Since it contains the elements of a simple flyball governor, it will be used as an introduction to the study of this class of governors.

When the pendulum revolves about the axis at a uniform speed, the ball remains at a constant distance *r* from the axis and at a constant distance h below the point c. This vertical distance h, from the point of suspension to the plane in which the center of the ball moves, is known as the height of the pendulum. If the speed of rotation is increased, the ball swings outwards and upwards, increasing r and decreasing h. Decreasing the speed has the opposite effect.

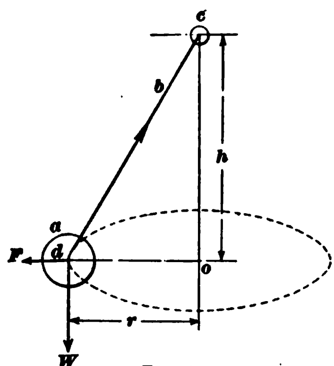


FIG. 4

7. While the ball is revolving at a constant rate, three forces are acting on it, namely, the tension in the string, the force of gravity, and the centrifugal force due to the rotation. These forces, at any definite speed, must be in equilibrium, since the ball remains at a constant distance from the axis at that speed.

Let F = centrifugal force, in pounds;
W = weight of ball, in pounds;
v = linear velocity of center of ball, in feet per second;
r = radius, in feet, of circle described by center of ball;
g = 32.16 = acceleration due to gravity.

Then, by the principles of mechanics,

$$F = \frac{Wv^2}{gr} \qquad (1)$$

The force F tends to throw the ball outwards, horizontally, in the direction of od. Gravity, acting with a force equal to W, the weight of the ball, tends to draw it vertically downwards. These two forces are held in equilibrium by the tension in the string. Let H be the height of the pendulum, in feet. Then, taking moments about c as a center, and remembering that, since the string tension acts through c, its moment is zero,

$$Wr = FH;$$

or

$$F = \frac{Wr}{H} \qquad (2)$$

Equating the values of F in formulas **1** and **2**,

$$\frac{Wv^2}{gr} = \frac{Wr}{H}$$

Solving for H, $$H = \frac{gr^2}{v^2} \qquad (3)$$

Let n represent the number of revolutions per minute of the ball. Then,

$$60\,v = 2\pi r n, \text{ or } \frac{r}{v} = \frac{30}{\pi n}, \text{ and } \frac{r^2}{v^2} = \frac{900}{\pi^2 n^2}$$

Substituting this value of $\frac{r^2}{v^2}$ in formula **3** and letting $h = 12\,H$, the height in inches,

$$h = \frac{12 \times 900\,g}{\pi^2 n^2} = \frac{12 \times 900 \times 32.16}{9.8696\,n^2} = \frac{35{,}192}{n^2}$$

The term 35,192 may be taken as 35,200 with very little error, and the formula then becomes

$$h = \frac{35{,}200}{n^2} \qquad (4)$$

EXAMPLE 1.—What is the height of a simple pendulum revolving 90 times per minute?

SOLUTION.—Applying formula **4**,

$$h = \frac{35{,}200}{90^2} = \frac{35{,}200}{8{,}100} = 4.35 \text{ in. Ans.}$$

EXAMPLE 2.—What is the speed of a simple revolving pendulum whose height is 5.5 inches?

SOLUTION.—Transposing formula **4**, $n = \sqrt{\frac{35{,}200}{h}}$. Substituting the value of h,

$$n = \sqrt{\frac{35{,}200}{5.5}} = \sqrt{6{,}400} = 80 \text{ rev. per min. Ans.}$$

8. An analysis of formula **4** shows that *the height of a revolving pendulum is independent of the weight of the ball and the length of the string, and depends solely on the number of revolutions.*

When the length of the string is less than the calculated height for a given speed, the ball will not rise, but will remain near the axis until the speed is increased sufficiently to give a value of h less than the length of the string. Table I has been calculated by the use of formula **4**.

It is evident, from a study of this table, that the change in height for a small change in speed is considerable at low rotative speeds, and that it is very small at high speeds. This is shown graphically by Fig. 5, which represents a

TABLE 1

GOVERNOR HEIGHTS AT VARIOUS SPEEDS

n, in Revolutions per Minute	h, in Inches	n, in Revolutions per Minute	h, in Inches
40	22.00	100	3.52
50	14.08	110	2.91
60	9.78	120	2.44
70	7.18	150	1.56
80	5.50	200	.88
90	4.35	300	.39

conical pendulum with an arm ab 12 inches long. In this case, the ball will hang against the spindle until the speed is great enough to make the value of h a little less than 12 inches. This, it will be seen from the table, occurs at a point between 50 and 60 revolutions per minute. At 60 revolutions per minute, the ball will be at c; and as the speed increases, it will rise successively to the positions on the arc $d\,b\,e$ corresponding to the number of revolutions per minute given on the vertical scale at the right.

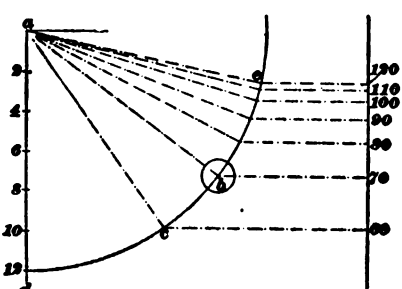


FIG. 5

9. The Simple Pendulum Governor.—The simple form of pendulum governor shown in Fig. 1 consists of two equal and opposed pendulums. In the simple revolving pendulum described in Art. 6, the string is assumed to have no appreciable weight. In the actual governor, the arm, corresponding to the string, has weight, and therefore the formulas for the simple revolving pendulum cannot be expected to apply accurately to the flyball governor. They do, however, form an easy means of approaching the study of the forces involved, and may be taken as being approximately correct for the simple pendulum governor.

10. In the simple form of flyball governor shown in Fig. 1, it will be seen that the forces acting on each ball (neglecting the weight of the arms *c* and *e* and their centrifugal force) are those that act on the revolving pendulum of Fig. 4—gravity, centrifugal force, and the tension in the arm. Hence, the action of this governor under changes of speed will be very much like the action of the simple revolving pendulum. That is, the height of the governor will vary considerably for small changes in speed when it is running slowly, and will vary but little when the speed is high.

The change of height of the governor furnishes the motion required to cause the regulation; and if this change of height is extremely small, the regulation cannot be properly effected. Suppose, for example, that a governor of this class is to run at a speed of not less than 110 and not more than 120 revolutions per minute. Such a governor, in order to regulate properly, should give a vertical movement of at least $1\frac{1}{2}$ inches to the sleeve throughout its full range. But, according to Table I, a change of height of only about $\frac{1}{2}$ inch can be expected; hence the unsuitability of this simple type for governing at high speeds.

If, however, a weight is added to the sleeve, so that there is a greater downward pull on the balls without any increase in the centrifugal force, then, for a given speed, the balls will be lower and the height greater than in the simple unloaded type, and, consequently, the change in height for a definite change of speed will be increased. Thus, by loading the sleeve with a weight, the governor becomes more valuable for regulation at the higher speeds.

11. The Loaded Pendulum Governor.—A form of loaded governor, known as the **Porter governor,** is illustrated in Fig. 6. The balls *a, a* are attached to the arms b, b pivoted to the spindle c, and are connected by the arms d, d to a weight e fastened to the sleeve f. Both the weight and the sleeve are free to move up or down along the spindle. The arms b and d are assumed to be equal in length, and as a result the weight e rises twice as far as the balls, for a given rise of the balls. Letting L represent the weight of the center load, it is evident that $\frac{1}{2}L$ acts on each ball. But since, for a given movement, the weight rises twice as far as the balls, it follows that the work done in raising the weight is equivalent to raising twice the weight through one-half the distance; that is, it is equivalent to raising $2 \times \frac{1}{2}L = L$ to the height through which the governor ball rises, the effect being the same as though a downward force equal to L acted directly on each ball.

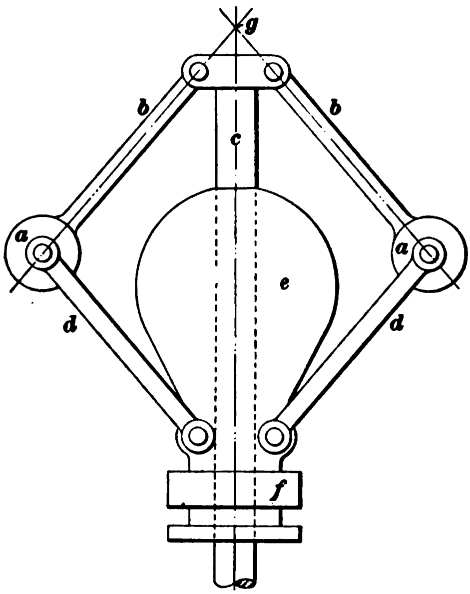


FIG. 6

Let F = centrifugal force of one ball, in pounds;
H = height of governor, in feet;
r = radius of circle described by center of ball, in feet;
W = weight of one ball, in pounds;
L = weight of center load, in pounds;
v = linear velocity of center of ball, in feet per second.

Then, equating the moments of the centrifugal and gravity forces, as in Art. **7**,

$$(W + L)r = FH;$$

or

$$F = \frac{(W + L)r}{H} \qquad (1)$$

But $F = \frac{W v^2}{g r}$, according to the formula for centrifugal force. Hence,

$$\frac{W v^2}{g r} = \frac{(W+L) r}{H};$$

or

$$H = \frac{g (W+L) r^2}{W v^2}$$

Taking $h = 12 H$ and $\frac{r^2}{v^2} = \frac{900}{\pi^2 n^2}$, as in Art. **7**,

$$h = \frac{12 g (W+L) 900}{W \pi^2 n^2} = \frac{W+L}{W} \times \frac{35{,}200}{n^2} \qquad (2)$$

12. If the arms b and d, Fig. 6, are not equal, the weight e will not move twice the distance through which the balls rise. In such a case, let K represent the ratio of the vertical rise of the weight to the corresponding vertical rise of the balls. Then, the formulas of the preceding article are modified somewhat, to conform to the new conditions, as follows:

$$\left(W + K\frac{L}{2}\right) r = F H;$$

or

$$F = \left(W + K\frac{L}{2}\right)\frac{r}{H} \qquad (1)$$

By the course of reasoning already explained, the height of the governor is found to be

$$h = \frac{W + K\frac{L}{2}}{W} \times \frac{35{,}200}{n^2} \qquad (2)$$

If the value of K is taken as 2, it will be seen that these formulas become identical with those in the preceding article.

In Fig. 6, the arms b, b are not pivoted on the axis of the spindle, but at either side of it. Consequently, to find the height h, the center lines of the arms must be produced until they intersect the axis at g, which is the virtual point of suspension, and the height is then measured by the vertical distance from g to the plane of rotation of the ball centers.

EXAMPLE 1.—In a loaded governor having arms of equal length, the weight of each ball is 5 pounds, and the center weight is

30 pounds; what is the height of the governor at 200 revolutions per minute?

SOLUTION.—Apply formula **2**, Art. **11**, making $W = 5$, $L = 30$, and $n^2 = 40{,}000$. Then,

$$h = \frac{5 + 30}{5} \times \frac{35{,}200}{40{,}000} = \frac{35}{5} \times \frac{35{,}200}{40{,}000} = 6.16 \text{ in. Ans.}$$

EXAMPLE 2.—A loaded governor has arms of such length that the center weight rises $2\frac{1}{3}$ times as far as the balls; the center weight weighs 42 pounds and each ball weighs 7 pounds. Find the height of the governor at 180 revolutions per minute.

SOLUTION.—Apply formula **2**, Art. **12**. $W = 7$, $K = \frac{7}{3}$, $L = 42$, and $n^2 = 32{,}400$. Then, substituting these values,

$$h = \frac{7 + \frac{7}{3} \times \frac{42}{2}}{7} \times \frac{35{,}200}{32{,}400} = \frac{56}{7} \times \frac{35{,}200}{32{,}400} = 8.69 \text{ in. Ans.}$$

13. The Spring Loaded Governor.—In the loaded governor, the resistance to the outward motion of the balls is furnished by the downward forces due to the weight of the balls and of the center weight. This resistance, however, may be furnished by a spring, in which case the governor is said to be *spring loaded.* The **Hartnell governor,** shown in Fig. 7, is of this type. The balls a, a are fastened to the vertical arms of bell-crank levers b, b. These levers are pivoted at c, c to a rigid frame that turns with the spindle d. The horizontal arms of the bell-cranks carry rollers at their ends, against which a sliding collar e is pressed by the action of the helical spring f.

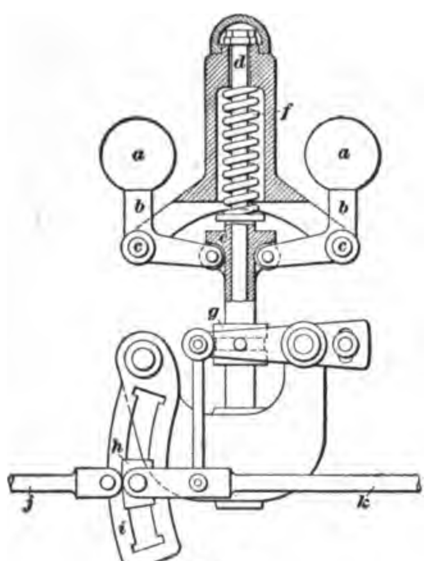


FIG. 7

When the governor revolves, the centrifugal force throws the balls outwards, and the bell-cranks, turning on the pivots c, c, raise the collar e and compress the spring until the moment of the spring pressure about the pivot c is equal to the moment of the centrifugal force about the same point. As the collar e rises, it carries the sleeve g with it, thus lifting the block h higher in the slotted link i. This link is given a to-and-fro motion by an eccentric attached to the rod j. Hence, as the block h is raised, the reciprocating movement of the rod k becomes shorter; this rod is attached to the valve stem, and the valve travel is therefore reduced and cut-off is made earlier.

14. Isochronism.—If a governor is so designed that its height is the same for all positions of the balls, the governor is said to be isochronous, and therefore possesses the quality of **isochronism**. But, by formula **4**, Art. **7**, $h = \frac{35,200}{n^2}$. Now, if h remains constant, it is evident that n^2, and consequently n, must remain constant. Hence, an isochronous governor also has the same speed for all positions of the balls. Since the speed of the governor is directly dependent on and proportional to the speed of the engine, it follows that the engine speed must be constant for all positions of the governor balls.

An isochronous governor, or a governor possessing isochronism, is therefore one that keeps the engine running at a uniform speed at all loads. To fulfil this condition, the moment of the centrifugal force must equal the moment of the spring or weight, in all positions, at a given speed of rotation of the governor. An absolutely isochronous governor is therefore in neutral equilibrium and is worthless, so far as speed regulation is concerned.

15. To illustrate, suppose an engine to be equipped with an isochronous governor. If the load is somewhat reduced, the engine speed will increase slightly, and the increased centrifugal force due to the higher speed will cause the balls to move further from the spindle. Before the change of speed, the moment of the centrifugal force was equal to that of the spring or weight. But now that the speed is increased, the centrifugal moment is the greater, and remains so, and hence the balls continue to move outwards under the action of this force until the extreme outward position is reached, and the admission valve or throttling valve is nearly or completely closed. As soon as this occurs, the engine will slow down, and at the reduced speed the moment of the centrifugal force will become less than that of the spring or weight, and as a result the balls will fall to their extreme inner position, opening the admission valve fully. This will cause a sudden increase of speed, and the action just described will be repeated indefinitely. This alternate speeding up and slowing down of the engine is termed **hunting.**

It is desirable to have an engine run at nearly the same speed at all loads, and therefore a governor should be as nearly isochronous as possible; but it is also necessary that the governor shall maintain a definite position for a given speed. For this reason, the best governors are not entirely isochronous.

16. Static and Astatic Governors.—A governor is said to be **static** when, at a given speed, it keeps a definite equilibrium position in which the moment of the centrifugal force and the moment of the resistance are equal. If the balls of a static governor are displaced from this equilibrium position, they will tend to move back to it at once. This means that the moment of the resistance is greater than the moment of the centrifugal force at all positions above the equilibrium position, and less than the centrifugal moment at all positions below the equilibrium position.

On the other hand, if the reverse is true, on displacing the ball outwards from the equilibrium position the centrifugal moment increases more rapidly than the resisting moment, and the balls fly to the outer extreme. If displaced inwardly from the equilibrium position, the moment of the resistance, due to the spring or to the load, will overcome the centrifugal moment and draw the balls to the extreme inner position. Such a governor is said to be **astatic.**

17. Summary.—Three kinds of governor action have been explained in the foregoing paragraphs. In order that they may be fixed clearly in mind, the general characteristics of each are here repeated.

1. *Isochronous.*—The governor, if displaced from its equilibrium position, alternately swings from one to the other of its extreme positions, and keeps up this action indefinitely.

2. *Static.*—The governor, if displaced from its equilibrium position, tends to return to that position.

3. *Astatic.*—The governor, if displaced from its equilibrium position, at once moves to either the inner or outer extreme position and remains there.

18. Stability.—For any governor position, there is a definite corresponding power of the engine, other conditions remaining the same. The heaviest load, or the maximum power, corresponds to the position of the balls nearest the spindle. With a change of load, there is a change of position of the flyballs. But they do not move directly from one position to the other and stay there. Usually, the balls swing from the old position past the new, and then back toward the old, oscillating thus until they come to rest.

Now, one of two things will occur: (1) The oscillations past the new position will grow less and less and finally die out, leaving the governor in the proper new position. (2) The oscillations will increase until the governor reaches its extreme limiting position.

In the first case, the governor is **stable**, or possesses **stability**; in the second case, it is **unstable.** It is evident that an unstable governor is useless and that stability is a necessary condition of a good governor.

19. Coefficient of Speed Variation.—If the difference between the highest and lowest speeds fixed by the governor is divided by the mean speed, the quotient will be the **coefficient of speed variation.** For example, if an engine has a mean speed of 250 revolutions per minute, while at the extreme governor positions the speeds are 247 and 253 revolutions, the coefficient of speed variation is

$$\frac{253 - 247}{250} = .024, \text{ or } 2.4 \text{ per cent.}$$

Let k = coefficient of speed variation;

n_1 = maximum number of revolutions per minute;

n_2 = minimum number of revolutions per minute;

n = mean number of revolutions per minute.

Then, $$k = \frac{n_1 - n_2}{n}$$

20. Fundamental Equation for Stability.—The stability of the pendulum governor depends on the valuc of the coefficient of speed variation. The least value of k that may be used if a stable governor is required is given by the following formula, which is recommended by well-known authorities on governors:

$$k = 4{,}000\sqrt[3]{\frac{G a}{F}} \times \sqrt[3]{\frac{H^2}{W^2 N^4 R^4}} \qquad (1)$$

in which k = least coefficient of speed variation;

G = reduced weight of governor, which will be explained later;

a = horizontal movement of flyballs, in feet;

F = mean centrifugal force of balls, in pounds;

H = indicated horsepower of engine;

W = weight of flywheel, in pounds;

N = number of revolutions per minute of main shaft;

R = radius of flywheel, in feet.

It is desirable that the speed variation between heavy and light loads should be small. It has been found, however, that the smallest value of k given by the foregoing formula will not give the best regulation, and that the best value of k is double that obtained by the formula.

The reduced weight of the governor, denoted by the symbol G, is that weight which, if concentrated at the center of gravity of the flyballs, will give the same effect as all the weights moving as they actually do. The value of G is found as follows:

Let G_1, G_2, G_3, etc. denote the weights of the several moving parts of the governor, such as balls, load, sleeve, etc.; S_1, S_2, S_3, etc., the distances through which these weights move from one position to the other; and a the movement of the flyballs at right angles to the spindle. Then,

$$G = \frac{G_1 S_1^2 + G_2 S_2^2 + G_3 S_3^2 + \text{etc.}}{a^2} \qquad (2)$$

SHAFT GOVERNORS

21. Centrifugal Shaft Governor.—A form of centrifugal shaft governor is shown in Fig. 8. The eccentric a is

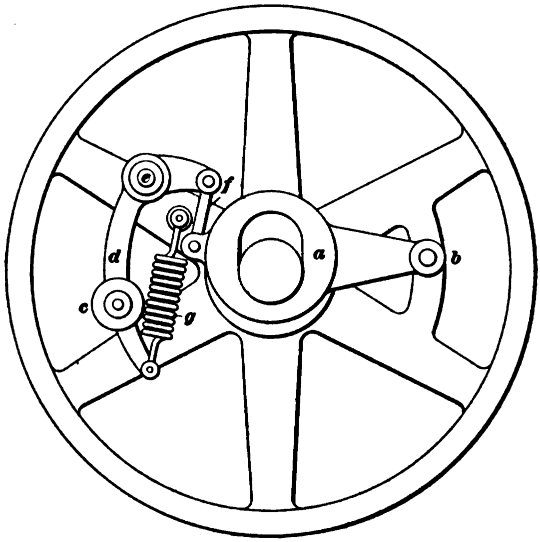


FIG. 8

pivoted at b, and as a consequence the center of the eccentric swings across the shaft in an arc whose center is at b, the slot in the eccentric permitting this motion. The movement of the eccentric is due to the action of centrifugal force on the weight c attached to the arm d pivoted at e. As the speed increases, c moves outwards, the arm d turns on e and the link f forces the eccentric across the shaft, bringing the eccentric center nearer the shaft center, decreasing the throw and the valve travel, and thus causing earlier cut-off. If the speed decreases, the spring g pulls the arm d inwards and the cut-off is made later. The regulating force is thus dependent on centrifugal force.

22. Inertia Shaft Governor.—If the inertia of a weight is employed, as well as the centrifugal effect, the governing action is much hastened. In Fig. 9 is shown a **Rites inertia governor,** so named from its inventor, F. M. Rites. It combines the centrifugal and inertia forces to produce speed regulation. Two weights *a*, *a* are fastened to a bar that is pivoted at *O*. This bar, with the weights, is

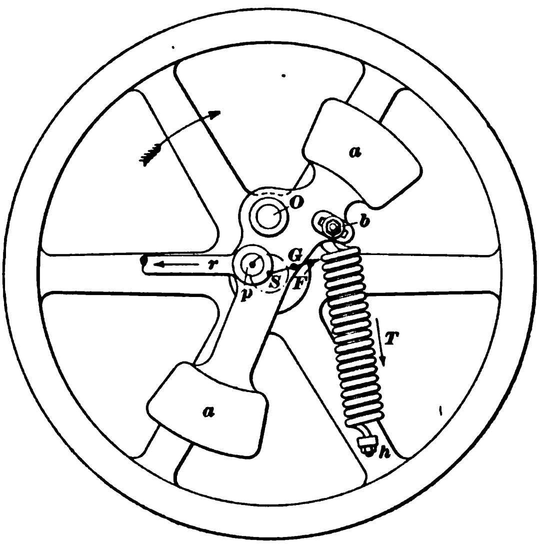


FIG. 9

called the weight arm and swings on the pin at *O*, this pin being attached to the flywheel. The center of the shaft is at *S* and the center of gravity of the weight arm at *G*. The valve rod *r* is connected to the weight arm at the point *p*, thus obviating the use of an eccentric. A spring is attached to the weight arm at *b* and to the wheel at *h*.

23. The external forces acting on the weight arm are three in number: (1) The pull *T* of the spring; (2) the pull or thrust of the valve rod; and (3) the pressure on the pin *O*. Since the pressure on *O* acts through the center of the pin, it has no turning moment about O and consequently cannot have any effect in swinging the weight arm on its pivot, disregarding the effect of friction on the pin. The only force within the weight arm that can produce motion is the centrifugal force F, which acts through G radially outwards from S.

Taking O as the center of moments, the internal and external forces acting on the weight arm produce three turning moments. These are: (1) The moment of the centrifugal force F; (2) the moment of the spring pull T; and (3) the moment of the valve-rod pull or thrust. When the weight arm is not turning on the pin O, these moments must be in equilibrium and their algebraic sum must be equal to zero. The moments of these three forces may be readily calculated.

Let W = weight of weight arm, in pounds;
v = velocity of point G, in feet per second;
r = radius SG, in feet;
g = 32.16;
x = perpendicular distance from O to line SG.

Then, $F = \frac{Wv^2}{gr}$ and the moment M_c of the centrifugal force, in inch-pounds, is Fx. Hence,

$$M_c = Fx = \frac{Wv^2x}{gr} \qquad (1)$$

The moment M_s of the spring pull, in inch-pounds, is found as follows:

Let s = pull, in pounds, required to produce 1 inch of extension of spring;
y = extension of spring, in inches;
d = perpendicular distance from O to line of action of spring pull.

Since the pull of the spring is directly proportional to its extension, the pull for any extension y is sy, and

$$M_s = syd \qquad (2)$$

24. In most shaft governors, the eccentric is attached directly to the weight arm, so that the pull or thrust of the valve rod is transmitted directly to the weight arm. It is largely due to this fact that the weight arm must be made heavy. The force exerted by the valve rod is quite variable, since it is the resultant of a number of forces. These are: (1) The valve resistance, including the friction of the valve-stem stuffingbox; (2) the force required to accelerate the valve, valve rod, valve stem, etc.; (3) the unbalanced steam pressure on the valve stem; (4) the weight of the parts, in the case of vertical engines; (5) the centrifugal force of the valve rod and eccentric, where an eccentric is used. Each of these forces can be calculated more or less accurately, and their resultant at any moment may be found. The moment M_v of these forces is then equal to the product of their resultant and the perpendicular distance in inches from O to the line of action of the resultant.

Then, $$M_c + M_s + M_v = 0 \qquad (3)$$

That is, the algebraic sum of the moments due to the centrifugal force, the spring pull, and the valve rod pull is equal to zero.

25. The position of the center of gravity G of the weight arm, Fig. 9, is very important in securing the proper inertia action of the governor. In order that the inertia action shall be beneficial to the governor, the center of gravity G of the weight arm should always lie on the side of OS toward which O is moving. Preferably, G should lie outside a semicircle drawn on OS as a diameter, but in some cases G may be on the circumference, or a little inside.

The stability of the shaft governor depends primarily on three things: (1) The coefficient of speed variation; (2) the inertia action of the governor; (3) the friction in the governor mechanism.

It may be stated in the first place that stability cannot be had in an inertia governor unless there is some friction present to damp or resist the oscillations of the weight arm when the governor is moved in obedience to a change of load. Usually, there is sufficient friction in the weight-arm pin and in the various joints of the valve gear attached to the weight arm; but, if there is not enough friction for the purpose, the governor must be provided with a dashpot.

26. Relative Advantages of Centrifugal and Inertia Governors.—The following general statements may be made regarding the relative advantages of the different types of governors:

A purely centrifugal governor cannot be stable without a considerable speed variation from heavy to light load. An inertia governor, on the other hand, may be made nearly isochronous if desired.

The inertia type is better adapted to engines subjected to considerable and sudden changes of load. In the case of an engine running continuously at about the same load, the inertia action would be small, and the addition to the weight in order to secure a large enough inertia moment might, under such conditions, hinder instead of increase the promptness of the governing action. In such a case it is probably wise to choose a centrifugal governor.

To secure the best regulation by an inertia governor, the valve-gear pull should act directly on the weight arm. Balanced valves should be used, and the weight arm must be heavy and have a large inertia moment. If a balanced valve is not used, the arrangement must be such that the pull of the valve rod will influence the action of the governor as little as possible.

GOVERNOR CALCULATIONS

THE FLYBALL GOVERNOR

27. Design of the Common Flyball Governor.—The first step in the design of a flyball governor is to find

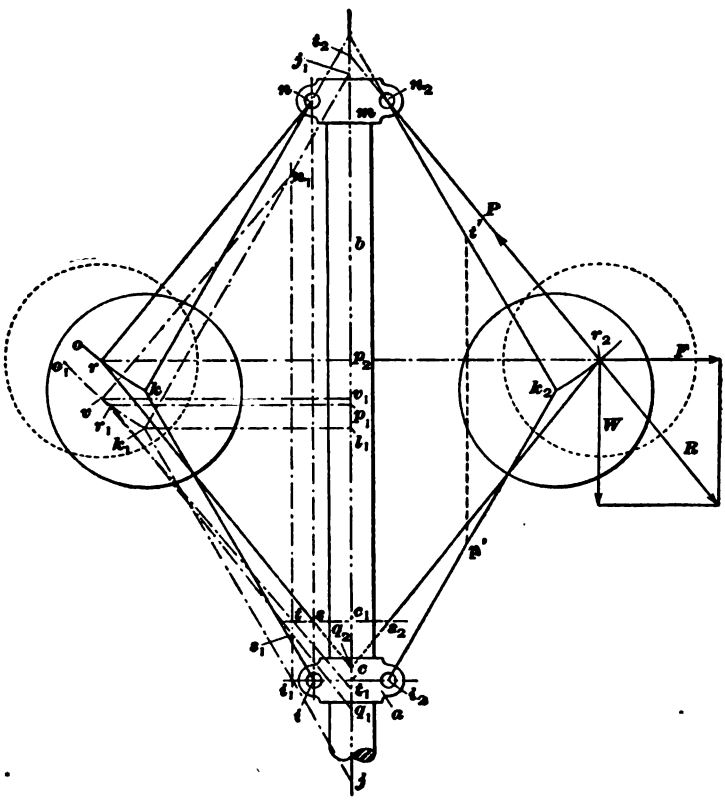


FIG. 10

the required movement of the sliding collar or sleeve on the spindle. By the time the design of the governor is reached, the design of the engine will be for the most part completed, and so the position of the governor as well as the dimensions of the links connecting it to the valve gear will be known. On the drawing board, therefore, lay out the position of the governor sleeve when the engine is taking the maximum amount of steam per stroke. Then lay out the position of the sleeve when the steam is entirely shut off from the cylinder. The two positions thus found will be the lowest and highest positions, respectively, of the sliding collar, and the distance between them will represent the required movement of the collar on the spindle. In Fig. 10, which represents the layout of a Corliss engine governor, these positions are at c and c_1, respectively.

The minimum coefficient of speed variation may be found by formula **1**, Art. **20**. Assume this value, in the present example, to be .035. Then, for the best operation, twice the minimum value determined by this formula, or .07, should be taken. If, however, the coefficient of speed variation is indicated by the specifications under which an engine is to be built, that coefficient should be used instead of twice the calculated value. The allowable speed variation is usually divided equally above and below the mean speed. Thus, if the Corliss engine of this problem is to have a mean speed of 75 revolutions per minute, the greatest speed will be

$$75 + \left(\frac{.07}{2} \times 75\right) = 77.6 \text{ revolutions per minute}$$

and the least speed will be

$$75 - \left(\frac{.07}{2} \times 75\right) = 72.4 \text{ revolutions per minute}$$

The allowable variation of speed between no load and full load must be determined for each case.

28. Frictional Resistance.—The **frictional resistance** of the sliding collar cannot be determined accurately. The most satisfactory way is to give it a value obtained from tests of engines of similar design. It will probably vary from a fraction of a pound in a governor actuating a small, well-balanced throttling valve, to 3 pounds in a governor actuating a series of links, with an average value of about 2 pounds. If in doubt, the safe plan is to assume a large value, resulting in a heavier governor.

29. Sudden Variation of Speed.—It is necessary to decide next how much variation of speed to allow, on a sudden change of load, before the governor begins to respond. Experiments show that this may be as much as 5 per cent., and even more, in actual running; but it is better not to allow more than 3 per cent. in design—if possible, not more than 2 per cent. The greater the frictional resistance to movement of the governor, the greater is the variation of speed that must be allowed on sudden change of load.

Let W = weight of one governor ball, in pounds;
C = frictional resistance to movement of sliding collar, in pounds;
D = fraction by which speed must change to set sliding collar in motion.

Then, assuming that the arms of the governor shown in Fig. 10 are of the same length, which is the usual design,

$$W = \frac{C}{2D}$$

This formula is based on a series of mathematical processes that are not given here owing to their length and intricacy. Using the data of the preceding article, the value of C is taken as 1.75 pounds, which insures a powerful governor, since actual tests of similar existing gears gave C a value of only about 1.5 pounds.

With this value of C, it has been found that the variation of speed on change of load may be kept down to about 2.5 per cent.; that is, $D = .025$. If, with the sliding collar clear down, some of the load is thrown off, the speed will go up to $72.4 \times 1.025 = 74.21$ revolutions per minute before the governor begins to act. The balls will then rise to a new equilibrium position and will remain there as long as the engine continues to make 74.21 revolutions per minute. Then, by the preceding formula,

$$W = \frac{1.75}{2 \times .025} = 35 \text{ pounds}$$

If the balls are of cast iron, each must be approximately $6\frac{5}{8}$ inches in diameter to give the required weight.

30. With the balls in their lowest position, there should be at least $1\frac{1}{2}$ inches of clearance between the spindle and each ball. A diameter of 3 inches should be allowed for the diameter of the spindle *b*, Fig. 10. This is the maximum, and may be considerably reduced after the design is nearly completed, but no greater value than this will be needed. The angle between the spindle and the arms in the lowest position is taken as 30°. Laying out these dimensions to scale on the drawing board, the height of the governor is found to be 10.75 inches by actual measurement. Substituting this value for h in formula **4**, Art. **7**, and solving for n,

$$n = \sqrt{\frac{35{,}200}{10.75}} = 57.22$$

That is, the governor makes 57.22 revolutions per minute with the engine running at 72.4 revolutions per minute. Hence, the ratio of the governor speed to the engine speed is $\frac{57.22}{72.4} = .79$. For convenience, this may be taken as $\frac{3}{4}$, and the ratio of the diameter of the governor pulley to that of the pulley on the main shaft is made 4 to 3. Then, the speed of the governor in its extreme low position is

$$n = 72.4 \times \frac{3}{4} = 54.3 \text{ revolutions per minute}$$

and the height of the governor is

$$h = 35{,}200 \div 54.3^2 = 11.94 \text{ inches}$$

31. In Fig. 10, the point at which the sliding collar *a* is attached to the lower arm of the governor, when the latter is in its lowest position, must lie on a horizontal line through the point *c*. As a guide, it may be said that, with the regulation desired in this case, this point may be expected to come well to the left of *c*. With closer regulation, the point would come closer to *c*; while with very close regulation, it would come on the opposite side of *c* from the ball to which it is connected, which makes what is called a cross-arm governor. As a trial position of this joint, the point i_1 is located 2 inches to the left of the point c. Then through i_1 a line $j i_1 k_1$ is drawn so as to make an angle of 30° with the axis of the spindle. From the intersection j on the spindle, the height of the governor at full load, 11.94 inches, is laid off upwards to l_1, and a horizontal line through l_1 intersecting $j i_1 k_1$ at k_1 locates the center of the governor ball at full load.

The height of the governor is laid off upwards from j merely as a matter of convenience. For, if $k_1 j_1$ is drawn so as to make an angle of 30° with the axis of the spindle, it is found that $j_1 l_1$ is equal to $j l_1$. But as the arms are equal, $j_1 l_1$ represents correctly the height of the governor at no load, and hence $j l_1$ also is the correct height.

32. The arms being equal in length, the joint between the fixed upper collar m and the upper arm must lie at n_1, vertically above i_1, and at a distance $k_1 n_1 = k_1 i_1$ from k_1. As the speed of the governor increases, the ball swings outwards, about n_1 as a center, along the arc $k_1 o_1$. Hence, at the highest position of the balls, the joint between the sliding collar and the lower arm must be taken at a point t on the vertical line $i_1 n_1$ at the same height as c_1, while the joint between the upper collar and the upper arm remains at n_1. With a radius equal to $k_1 i_1$ and with t as a center, strike an arc intersecting the arc $k_1 o_1$ at v; then v represents the center of the ball when the sliding collar is at c_1, corresponding to the no-load position. Drawing the horizontal line $v v_1$ and extending $v t$ to intersect the axis of the spindle at t_1, the height of the governor at no load is represented by the distance $t_1 v_1$.

The speed of the engine at no load being 77.6 revolutions per minute, that of the governor is

$$n = 77.6 \times \tfrac{3}{4} = 58.2$$

and the height of the governor at no load is

$$h = \frac{35{,}200}{58.2^2} = 10.4 \text{ inches, nearly}$$

But the distance $t_1 v_1$ measures considerably less than 10.4 inches, so that the governor will not give the desired height at no load. Furthermore, when the height of the governor is 10.4 inches, as shown at $q_1 p_1$, with the center of the ball at r_1, the joint between the sliding collar and the lower arm will be at s_1, where $r_1 q_1$ intersects $i_1 n_1$. So that even with the proper variation in height from no load to full load, the governor will not give the proper amount of motion to the sliding collar. As so far designed, therefore, the governor is unsatisfactory, since it will not raise the sliding collar to the required height at the maximum speed. Consequently, a new trial must be made.

33. The sliding collar must be raised to the required height without exceeding the desired change in the height of the governor. To accomplish this, the governor must move the sliding collar farther, for a given motion of the balls, which may be done by moving the points of attachment of the arms to the collars closer to the axis of the spindle. If the first assumption had given the sliding collar too much motion, an outward movement of the arm and collar joints would have been necessary.

34. Successive positions of the joint between the sliding collar and the lower arm, all on the line $c i_1$, must now be tried. For each new position of the joint, the following steps must be taken: (1) Find the position of the center of the ball with the sliding collar in its lowest position; (2) find the position of the joint between the upper arm and the upper collar; (3) find the position of the center of the ball with the middle line of the sliding collar at the height c_1; (4) measure the resulting height of the governor; (5) compare this height with the height corresponding to the desired number of revolutions at no load. The method of taking these steps is a repetition of that given in detail for the first trial.

In the example, the joint between the sliding collar and the lower arm was gradually moved inwards toward c, the height of the governor in the no-load or full-speed position being greater with each successive position of the joint, until finally a solution was reached with the center of the joint at i. This puts the center of the ball at k at full load and extreme low position of the governor balls, and the center of the joint between the upper arm and the upper collar at n. The ball moves in a circular arc $k o$ of radius $k n = k i$ about n as a center. The no-load or highest position of the center of the joint between the sliding collar and the lower arm is then at s, which puts the center of the ball at r. The resulting height of the governor, $p_1 q_1$, is found to be just the desired 10.4 inches.

35. It is now necessary to see what angle the arm makes with the horizontal, with the governor in its highest position, as the gear may cramp if this is too small. In the example, this angle is found to be about 50°, which may be allowed without hesitation. A glance at the drawing shows that there will be plenty of clearance between the spindle and the balls, with the latter in their lowest position.

It is also necessary to examine the drawing to see that an outward swing of the ball always produces a decrease in the height of the governor, as the governor will not work satisfactorily otherwise. This, in the example, is found to be correct. The right side of the governor may then be put in symmetrically with the left, as shown, the letters being the same as the corresponding ones on the left side, but having the subscript 2.

36. The pull in the upper arm $k_1 n_1$ is found by the following method: When the governor is revolving at a given speed and the friction of the collar on the spindle is neglected, the ball is held in equilibrium by three forces, namely, the centrifugal force F, the weight W of the ball, and the pull of the upper arm. These forces are shown in their relative positions and magnitudes in Fig. 10. The resultant of F and W is R, and in order to have equilibrium, the pull P of the upper arm must be equal and opposite to R. Now, $R = \sqrt{F^2 + W^2}$, and since W does not change, it is evident that R is greatest when F is greatest, that is, when the governor ball is in its highest position and moving at its maximum speed. The pull P is therefore likewise at a maximum when the governor ball is in the highest position.

Continue $r_2 n_2$ until it intersects the axis of the spindle in t_2, and draw the horizontal line $p_2 r_2$. Then the triangle $p_2 t_2 r_2$ correctly represents the triangle of forces acting on the ball, and $p_2 t_2 : t_2 r_2 = W : R = W : P$

Now, if $p_2 t_2$ is taken to represent the weight of the ball to a certain scale, $t_2 r_2$ represents the pull of the upper arm, to the same scale. That is,

$$P = \frac{t_2 r_2 \times W}{p_2 t_2} \qquad (1)$$

The downward thrust on the spindle is equal to the entire weight of the two balls with their upper and lower arms.

The thrust in the lower arm is found as follows: When the governor is at rest, the collar is in its lowest position, and the weight of the balls rests on the lower arms, causing a thrust in each. Let k_2 be the position of the governor ball when in its lowest position, and let $k_2 n_2$ and $k_2 i_2$ represent the upper and lower arms, respectively. Draw $p' t'$ between $k_2 n_2$ and $k_2 i_2$ parallel and equal to $p_2 t_2$, representing the weight of the ball. Then, $k_2 p' t'$ is the triangle of forces when the ball is in its lowest position, and the thrust T in the lower arm is given by the formula

$$T = \frac{k_2 p' \times W}{p' t'} \qquad (2)$$

There is no bending moment on the spindle, as the lateral forces are equal and opposed. All parts of the governor must be made amply strong to resist the forces as calculated by these formulas.

37. A useful modification of the common flyball governor is shown diagrammatically in Fig. 11. In the common construction of this governor, the sliding collar is at a. The arms $a\,b$ and $a\,b_1$ may be attached to the collar on the line of the axis of the spindle, or on either side of it, according to the closeness of regulation required, just as in the preceding governor. The same applies to the point of attachment of the arms $c\,d$ and $c\,d_1$ to the upper collar. The lower arms $a\,b$ and $a\,b_1$ are attached to the upper arms by pin joints at b and b_1, respectively. Usually, b comes in the middle of the length of cd, and b_1 in the middle of cd_1. Also, usually $ab = \frac{1}{2}cd$.

This form of governor is desirable when the frictional resistance of the sliding collar is large, or when the variation of speed on change of load must be kept small, since the governor balls may be made larger and the travel of the collar for a given rise of the balls may be kept small. The formulas in Art. 7 are applicable to this form of governor

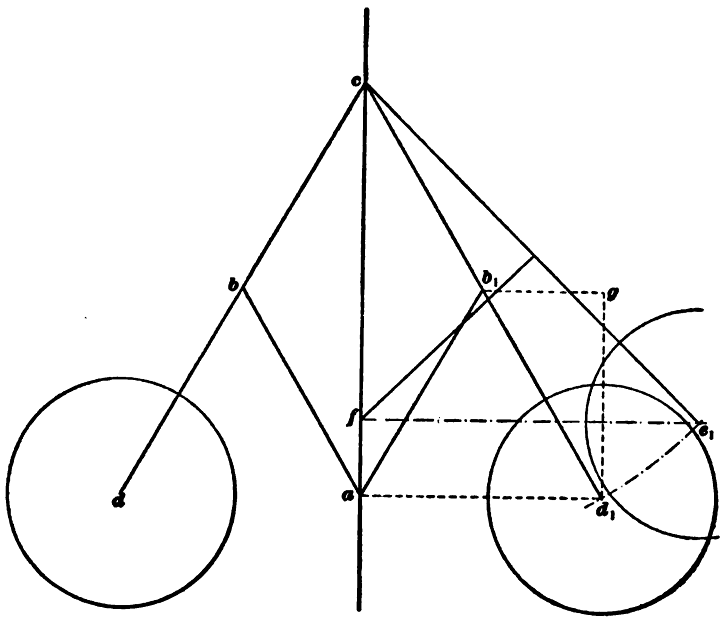


FIG. 11

also. However, it must be remembered that the height is the vertical distance between the plane of rotation of the ball centers and the point where the center lines of the upper arms meet the axis of the spindle.

Let W = weight of one ball of governor, in pounds;

D = fraction by which speed varies on change of load, before governor begins to respond;

C = frictional resistance to motion of sliding collar, in pounds.

Then, in the form of governor illustrated in Fig. 11,

$$W = \frac{C}{4\,D}$$

The weight of the balls being determined by this formula, the rest of the design can be carried out by the same general method that was employed in the design of the governor in Fig. 10. The same precautions should be taken as to the angle of the arms with the horizontal positions, and the decrease of governor height with outward swing of the balls.

38. To determine the maximum forces acting on the members of the governor, place the center of the ball in its highest position e_1, Fig. 11, and draw the horizontal line $e_1 f$. The height of the governor is then fc. Letting W represent the weight of one ball, the pull P on the upper arm is found by the formula

$$P = \frac{W \times e_1 c}{fc} \qquad (1)$$

When the governor is in its lowest position, with the center of the ball at d_1, there is a maximum bending moment M in the upper arm at b_1, due to the weight W of the ball. Draw $d_1 g$ vertically to meet a horizontal line through b_1 at g. Then, $b_1 g$ is the arm of the force W, and the bending moment is $M = W \times b_1 g$. Let ac represent the weight W to a certain scale. The triangles $a c d_1$ and $g d_1 b_1$ are similar, and so $b_1 g : a d_1 = b_1 d_1 : c d_1$, or

$$b_1 g = \frac{a d_1 \times b_1 d_1}{c d_1}$$

Substituting this value in the above expression for the bending moment,

$$M = \frac{W \times a d_1 \times b_1 d_1}{c d_1} \qquad (2)$$

The downward thrust on the spindle may be taken as the weight of the two balls with their arms. These formulas are not exact, but they give results that are sufficiently accurate for all practical purposes. The dimensions of the various parts must be calculated, by the principles of machine design, to resist the forces found by these formulas. A large factor of safety should be used. The short lower arms $a\,b$ and $a\,b_1$ may be made of the same diameter as the upper arms. It is not necessary to calculate the forces acting on them.

39. The Loaded Flyball Governor.—Where the frictional resistance to motion of the sliding collar of the governor is large, or where it is desired to run the governor at a high speed, or where it is necessary that the governor should be sensitive and respond to a change of load without allowing more than a small instantaneous variation of speed, the loaded flyball governor may be used. This governor, as most commonly constructed, was shown in Fig. 6. The load weight, which is shown inside the arms of the governor, surrounds the central spindle and is symmetrical with reference to it; it therefore has no centrifugal effect.

Let W = weight of one ball of governor, in pounds;
C = fraction by which speed will vary on change of load before governor begins to respond;
F = frictional resistance to motion of sliding collar, in pounds;
L = central load weight, in pounds;

Then,
$$C = 2(W+L)D \qquad (1)$$
Also, by formula **2**, Art. **11**,
$$h = \frac{W+L}{W} \times \frac{35{,}200}{n^2} \qquad (2)$$
These equations are based on the assumption that the arms of the governor are equal, this being the usual construction.

40. In designing this form of governor, it is first necessary to estimate a value of F that is sufficiently large, as in Art. **28.** A convenient value of h may then be assumed. The mean speed of the engine being known, its speed at full load may be determined exactly as in Art. **27.** For a trial value, the speed of the engine at full load may be taken as the value of n in formula **2**, Art. **39.** The variation of speed to be allowed on change of load must also be decided, as explained in Art. **29.** This determines the value of D.

The values of D, C, h, and n, thus determined, are then substituted in the formulas in Art. **39**, giving two equations containing only two unknown quantities, W and L. By solving these two equations, W and L become known. For, transposing the first equation,

$$W + L = \frac{C}{2D}$$

and transposing the second equation,

$$W + L = \frac{hWn^2}{35{,}200}$$

Equating these two values, $\frac{hWn^2}{35{,}200} = \frac{C}{2D}$. Solving for W,

$$W = \frac{35{,}200\,C}{2Dhn^2}$$

Having found W, its value may be substituted in either of the original equations and the value of L may thus be found.

41. Assuming an angle between the arms and the spindle of about 30° at full load, the results thus far obtained must be laid out to scale on the drawing board. If the governor arms do not allow ample room for balls and central load of the necessary weight, new assumptions must be made and the process repeated until the result is satisfactory. If desirable, n can be altered, in any ratio to the full-load speed of the engine, by interposing belting or gearing between the engine shaft and the spindle of the governor.

Having obtained satisfactory values of the height, speed, and weights, the points of attachment of the arms to the collar, so as to bring the sliding collar up to the proper point, when the governor has the height fixed by the no-load speed of the engine, must be determined on the drawing board by a trial process like that employed with the unloaded governor in Art. **34**.

42. It is customary to leave the upper part of the central load weight hollow, and put shot in the cavity to form a portion of the necessary weight. Openings are left in the top of the weight, through which shot can be taken out or put in. This is done in order that the governor may be adjusted under actual running conditions. It will be seen from the formulas in Art. **39** that adding shot and increasing L will increase the value of n for a given value of h; this will increase the speed at which governor and engine will run, and will also decrease the value of D for a given value of C; that is, it will increase the sensitiveness of the governor. The opposite effect may be produced by taking out shot and decreasing the value of L.

The pull in the upper arms and the thrust on the spindle may be obtained as in Art. **36.** The arms and spindle should be made amply strong to withstand the action of these forces. The lower arms may be made of the same diameter as the upper ones.

THE SHAFT GOVERNOR

43. Balancing the Gravity Effect.—In all shaft governors, gravity has a tendency to pull the weights downwards, and, unless this effect is balanced in some way, the action of the governor will be disturbed. In some governors, as in the case of the Buckeye governor, two weights are used, placed symmetrically on opposite sides of the shaft. The pull of gravity on one weight tending to turn the eccentric in one direction is exactly balanced by an equal pull on the other weight tending to turn it in the opposite direction. The governor is therefore balanced against gravity action at all positions.

In other governors, such as the straight-line engine governor, a single weight is used to balance the moment of the weight of the eccentric about its pivot. The latter method is preferable. For, if two springs are used on a two-weight governor, it will be found practically impossible to have their strengths exactly alike under varying tension; and if only one spring is used for both weights, it is necessary to transmit its force through links; the friction of the links, however, renders the governor less sensitive, which is undesirable. When the gravity moment of the eccentric is balanced by the gravity moment of one weight, the final result is generally better than where the gravity moment of one weight is balanced against that of another.

44. Effect of the Location of the Governor Weights. The governor members may be so disposed as to assist or obstruct, by their inertia, the proper response of the governor to change of load. Thus, in Fig. 12, let *a* be the center of the shaft about which the whole governor rotates; let *b* be

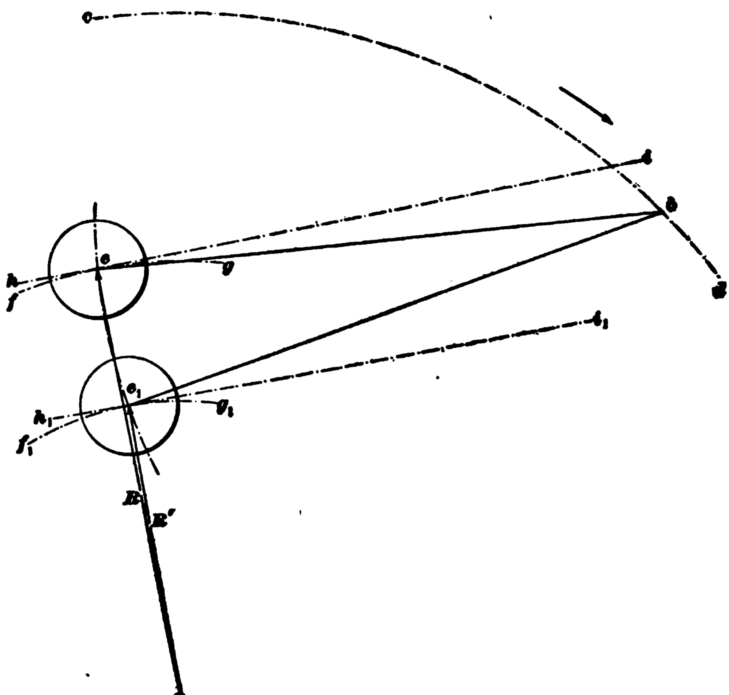


FIG. 12

the pivot of the weight arm, revolving about *a* in the circular path *c d*. The ball, whose center of gravity is at *e*, is attached to the pivot by the arm *b e*. The centrifugal force of the ball is supposed to be balanced by the tension in the governor spring, which, for the sake of simplicity, is not shown in the figure; so that, as long as the speed remains constant, the center of gravity of the ball remains at a constant distance from *a*. Then, the path of *e* is a circle *f g*, of radius *R*, about a as a center. When the speed decreases, the centrifugal force decreases, and the spring tends to draw the weight e in toward the center a; but, owing to its inertia, e tends to keep on traveling about a at the same angular velocity as before. This being greater now than the angular velocity of the point b, e exerts a pressure in the direction from h to i in the line $h\,i$, which is tangent to the path of e. This pressure has a moment about b that tends to move e away from a instead of toward it. When the speed increases, this moment is opposite and tends to move the weight toward a, thus being again opposed to the centrifugal force, which tends to move e away from a.

45. It is evident, therefore, that, with the ball at e, its inertia resists the proper action of the governor and will tend to make it sluggish, or slow to respond to change of speed. This is due to the fact that the tangent to $f\,g$ at e passes on the side of b away from a. If the ball is placed at e_1, so that $h_1\,i_1$, the tangent to its path at e_1, passes between b and a, it will be found that exactly the opposite occurs. With the ball at e_1, therefore, its inertia in its path $f_1 g_1$ assists the centrifugal force in producing the proper action of the governor, and tends to make the governor sensitive.

Therefore, in order that the inertia effects of the weights may be correct, the tangents to the paths of their centers of gravity at those centers should always pass between the center of the shaft and the center of the pivot of the weights. In this discussion, it is assumed that the weight follows its pivot in its rotation. When the direction of motion is such that the weight precedes the pivot, the conditions are reversed and the tangent to the path of the center of the weight, at its center, should lie outside the pivot point b.

46. The Rites Inertia Governor.—The use of the inertia to increase the sensitiveness of the governor is a marked characteristic of the type of governor known as the Rites inertia governor. The action of this governor does not differ in essential principles from that of the ordinary centrifugal shaft governor. In fact, its control of the speed really depends on the balance between the centrifugal force and the spring, the inertia effect simply increasing the sensitiveness of the governor.

47. The inertia governor is unbalanced against the action of gravity. This produces a tendency for the weight to continually rock back and forth, through a small arc on its pivot, as the wheel rotates, the spring preventing this arc from becoming large. Thus there is a tendency for the governor to work the eccentric in and out all the time through a small arc, the average position being that corresponding to the load. The effect of such motion of the eccentric will be to make the engine speed fluctuate all the time through a small cycle or wave, the mean speed of the engine being that desired. These fluctuations of speed, being small in amount, are not a serious matter where only the closest possible regulation to uniform mean speed under all conditions of load is desired, without the need of uniform angular velocity during each revolution. In such a case, especially where the fluctuations of load are sudden and violent, it is proper to use the inertia governor.

On the other hand, where the closest possible uniformity of angular velocity under given load and steam pressure is required, the lack of gravity balance in the inertia governor may prove a serious defect, and for such service the ordinary shaft governor, balanced against gravity action, is likely to give much better results.

48. Design of the Balanced Shaft Governor.—The design of the balanced shaft governor may best be explained by working out an actual example. Let it be required to design the governor for an 18″ × 30″ engine of the Buckeye type to run at a mean speed of 120 revolutions per minute. As in the case of the flyball governor, the problem must be worked out to a considerable extent, by a trial process, on the drawing board.

Draw the circle *a*, Fig. 13, representing the inner circumference of the governor-wheel rim, and the circle *b* to represent the shaft, choosing a suitable scale for the drawing.

Throughout the entire design, the crank is assumed to remain in the dead-center position *r s*. By means of the valve diagram, the path of the center of the cut-off eccentric, from the position of latest to the position of earliest cut-off, must be found. This path must then be laid off on the drawing.

In the figure, this is the arc *c d*, of about 90°, the eccentric

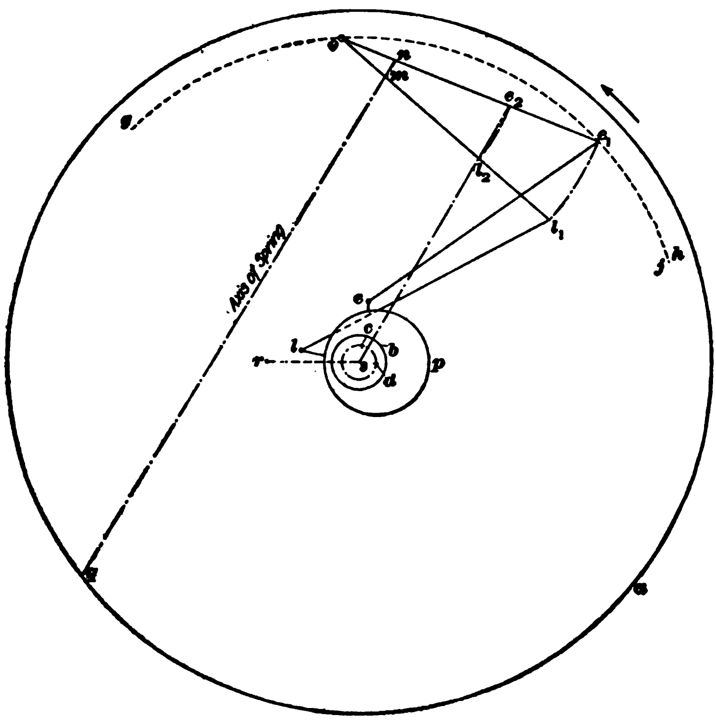


Fig. 13

center being at *c* to give the latest cut-off, and at *d*, directly opposite the crank-pin, to give the earliest cut-off. Draw the outer circumference *p* of the eccentric in any position. In Fig. 13, it is shown in the earliest cut-off position. On the eccentric, choose the positions of the pins connecting it to the links running to the weight arms of the governor. In the figure, these are at *l* and *e*, respectively, for latest and earliest cut-off. The rest of the design will deal with one side, or half, of the governor only, because, when it is completed, the other side can be put in symmetrically with it about the center of the shaft.

49. Taking up now the upper eccentric pin, as shown in the figure, it is plain that, with increase of speed, the governor must throw this pin from l to e, and that, with decrease of speed, it must throw it from e to l. Inside the inner circumference of the wheel, draw a circle $g h$ on the circumference of which the pivot for the weight arm of the governor is to be located, and a circle $i j$ whose circumference marks the outer limit of the center of the free end of the weight arm. These circles must be drawn inside the inner circle of the rim at distances great enough to allow a suitable fastening for the pivot of the weight arm and sufficient clearance for the weight. In this case, the two circles happen to be of the same radius, but this would not always be the case.

50. Next, by trial, find on the circumference $g h$ the point o at which the weight arm is pivoted; swinging on this point, the weight arm should, by a moderate motion inside the outer limit, shift the eccentric pin through the required arc without producing either too acute or too obtuse an angle between itself and the link connecting it to the eccentric pin, this link being connected to the free end of the weight arm. If the angle is too acute or too obtuse, the movement of the weight to produce the required movement of the eccentric will be excessive. The center so found must give a weight arm long enough, so that the weight and spring may be conveniently mounted on it and leave considerable room for adjustment. The latest cut-off position $o l_1$ of the weight arm, the link $l l_1$, connecting it to the eccentric pin, the earliest cut-off position $o e_1$ of the weight arm, and the link $e e_1$ connecting it to the eccentric pin are now drawn in.

At any convenient point l_2 on the arm $o l_1$, take a trial position of the center of gravity of the governor weight. The distance from the center of the shaft to l_2 can now be scaled off; in this case, it is found to be 2 feet $2\frac{1}{4}$ inches, or

2.1875 feet. There is now enough data to determine the size of the governor weights.

51. The weight required in the governor depends on the resistance to be overcome in operating the valve. This depends mostly on the inertia and friction of the valve. The inertia can be calculated; but the friction is always varying, depending on the tightness of stuffingboxes, condition of rubbing surfaces, tightness of balance plates, and quality of lubrication, so that no definite value can be assigned to it. The only course available, therefore, is to give the governor a moment of centrifugal force, which has been found by experience with similar designs to be sufficient. In the absence of exact data, a value of 16,000 inch-pounds, for the latest cut-off position, may be assumed and used in this case.

Let W = weight of each governor weight, in pounds;

N = number of revolutions of engine per minute;

R = radius of center of gravity of governor weight about center of shaft, in feet;

F_l = centrifugal force of weight, in latest cut-off position, in pounds.

Then, from the principles of mechanics,

$$F_l = .00034\, W N^2 R$$

52. The variation of speed to be assumed depends on the closeness of regulation for which the engine is to be designed. As in the case of the flyball governor, any desired value may be assigned to the variation of speed from no load to full load. This may be divided equally on each side of the mean speed, and the values of N in the extreme positions of the governor may thus be determined; but, in the case of the shaft governor, it is usually desired to attain the closest possible approach to isochronism. On the other hand, the greater the ratio in which no-load speed exceeds full-load speed, the greater will be the power of the governor to overcome any resistance that may oppose its operation and to resist any outside disturbing force that may tend to mar its action; that is, the greater is its stability.

The resistance to be overcome being to a considerable extent uncertain and the closest possible isochronism being desired, it is a good plan, in designing a shaft governor, to design for perfect isochronism, and to provide means by which the governor, after being erected on the engine, can be adjusted to as great a degree of stability as may be found necessary.

In the preceding formula, therefore, N will be taken as 120, the mean speed of the engine. R, as scaled off from the figure, is 2.1875 feet. Then,

$$F_l = .00034\, W \times (120)^2 \times 2.1875;$$

or

$$F_l = 10.71\, W$$

53. Draw a straight line from the center of the shaft through l, and drop a perpendicular from o, the center of the pivot of the weight arm, on this line. Scaling off the perpendicular distance from o to the line, it is found to be 19.625 inches; this is the arm of the centrifugal force about the center of the pivot. Let M_l be the moment of the centrifugal force of one weight at latest cut-off, and let M_e be the moment at earliest cut-off. Then,

$$M_l = 19.625 \times F_l = 19.625 \times 10.71\, W$$

But, according to Art. **51**, the total centrifugal moment F_l was taken as 16,000 inch-pounds, so that the centrifugal moment M_l of one weight is 8,000 pounds. Hence,

$$19.625 \times 10.71\, W = 8{,}000,$$

from which W = 38 pounds, very nearly.

54. The dimensions of the weight must now be found by trial. Assume the weight to be a cylinder 6 inches in diameter, counting the arm through the weight as part of the weight. The rest of the arm, and the difference in weight between cast iron and steel, may be neglected.

Let l = length of weight, in inches

Then $$38 = \frac{\pi}{4} \times 6^2 \times l \times \frac{450}{1{,}728},$$

from which $l = 5\frac{1}{4}$ inches, nearly.

By examination of Fig. 13, it is found that a weight of these dimensions may be put in, with its center of gravity at l_2, and still leave enough room for adjustment. The trial assumption of the position of the center of gravity of the weight is therefore satisfactory and may be retained.

55. At the position of earliest cut-off, the center of gravity of the weight will be at e_2. Scaling off from the drawing, its distance from the center of the shaft is now 2 feet 9 inches, or 2.75 feet. Then, by the method used in finding F_l, the centrifugal force of the weight at earliest cut-off is

$$F_e = .00034 \times 38 \times (120)^2 \times 2.75 = 512 \text{ pounds, nearly}$$

Scaling off the arm of this force about the pivot o, it is found to be 19.625 inches, as before. Then the moment of the centrifugal force at earliest cut-off is

$$M_e = 512 \times 19.625 = 10{,}048 \text{ inch-pounds}$$

56. A trial position of the point of attachment of the spring to the weight arm must now be assumed. In the figure, this is taken as m in the latest cut-off position. Then, at the earliest cut-off position, it will be at n; for intermediate positions, it will lie on the arc $m\,n$. In this case, a line may be drawn through m and n parallel to the line l_2e_2 that passes through the center of the shaft. By taking this line as the mean line of the axis of the spring and attaching the spring to the rim of the wheel at the point q, the moment of the spring pull about o is caused to vary in the same ratio as the moment of the centrifugal force of the weight about o, so that by balancing these two moments at the extreme positions, they will be balanced at all positions of the governor at the desired speed of rotation.

57. Now take the governor in its latest cut-off position. Draw the line $m\,q$ of the axis of the spring for this position of the governor. Scale off the perpendicular distance from o to $m\,q$, which, in this case, is found to be $6\frac{1}{4}$ inches; this is the arm of the spring pull about o, in the latest cut-off position. The governor is to remain in its extreme inner position until the speed is 120 revolutions per minute, and if the speed should exceed this the weight is to begin to move out. Therefore, the moment of spring pull about o, with the governor in its latest cut-off position, must equal the moment of the centrifugal force of the weight about o, with the governor in its latest cut-off position and at a speed of 120 revolutions per minute.

Let S_l equal the spring pull, in pounds, with the governor in the position of latest cut-off. Then, as the moment of the centrifugal force of one weight was taken as 8,000 inch-pounds,

$$S_l \times 6\tfrac{1}{4} = 8{,}000, \text{ or } S_l = 1{,}280 \text{ pounds}$$

Let S_e be the spring pull, in pounds, with the governor in the position of earliest cut-off. Then, scaling off the arm of this force about o, it is found to be $6\frac{1}{4}$ inches. Since the moments of the centrifugal force and spring pull must be equal,

$$S_e \times 6\tfrac{1}{4} = 10{,}048, \text{ or } S_e = 1{,}608 \text{ pounds}$$

58. From the study of machine design, it will be found that the elongation of a spring is directly proportional to the load applied to it. That is, if a given load produces a certain elongation, twice that load will produce twice the elongation, three times the load will produce three times the elongation, and so on. Or, if a given elongation is produced by a certain load, any fraction of the elongation will be produced by the same fraction of the load. Thus, if in a given spring an elongation of 4 inches is produced by a load of 1,000 pounds, an elongation of 1 inch will be produced by a load of one-fourth of 1,000 pounds, or 250 pounds, and an elongation of 2 inches by a load of 500 pounds.

59. Measuring on the drawing, it is found that the spring must elongate a distance $m\,n$, or $2\frac{1}{8}$ inches, from earliest to latest cut-off position. The load producing this elongation is the difference between the centrifugal forces in the two cut-off positions, or $1{,}608 - 1{,}280 = 328$ pounds. The load required to produce an elongation of 1 inch is therefore $328 \div 2\frac{1}{8} = 154.4$ pounds.

Since the spring in its latest cut-off position is to have an initial tension of 1,280 pounds, it must therefore have an initial elongation of $1{,}280 \div 154.4 = 8.29$ inches. Then, in the earliest cut-off position, the spring will be stretched $8.29 + 2.125 = 10.415$ inches.

60. It must now be seen whether a suitable spring to meet these requirements can be put in the space available for it on the governor wheel, the spring being proportioned according to the principles of machine design. The design, however, is largely a matter of trial, since the formulas do not give all the necessary proportions directly.

Round wire, being cheaper than square, should be used. A trial may therefore be made with a helical spring of $\frac{5}{8}$-inch round wire, and $2\frac{1}{2}$ inches mean diameter of coil. The mean diameter of the coil is twice the perpendicular distance from the center of the wire to the axis of the spring.

The formula for the greatest safe load of a helical spring is

$$W = \frac{.3927\ S d^3}{D}$$

in which W = safe load on spring, in pounds;
S = a constant, which may be taken as 50,000 for steel wire;
d = diameter of wire, in inches;
D = mean diameter of coil, in inches.

Substituting in the formula,

$$W = \frac{.3927 \times 50{,}000 \times (\frac{5}{8})^3}{2\frac{1}{2}} = 1{,}917.5 \text{ pounds}$$

The spring assumed will therefore carry the load. It now remains to be seen whether or not it will go into the space available on the wheel.

61. Since the elongation is proportional to the load, the required spring, when loaded to its full capacity, will stretch $1{,}917.5 \div 154.4 = 12.4$ inches, nearly. To find the elongation of a helical spring, the following formula should be used:

$$s = \frac{8\ W D^3 n}{G d^4}$$

in which s = elongation of spring, in inches;
W = load on spring, in pounds;
D = mean diameter of coil, in inches;
n = number of active coils in spring;
G = a constant = 12,000,000 for steel springs;
d = diameter of wire, in inches.

Substituting the known values in the foregoing formula,

$$s = 12.4 = \frac{8 \times 1{,}917.5 \times (2\tfrac{1}{2})^3 \times n}{12{,}000{,}000 \times (\tfrac{5}{8})^4}$$

from which $n = 95$ coils, nearly.

Then, under zero load, with the coils touching each other, the length of the active part of the spring will be $\frac{5}{8} \times 95 = 59\frac{3}{8}$ inches.

The length at latest cut-off will be the length under no load plus the initial elongation, or $59.375 + 8.29 = 67.7$ inches, very nearly.

Measurement on the drawing, Fig. 13, shows that there is not so much room as this available. The spring assumed will therefore not answer, and another trial must be made. The first trial, however, will serve as a guide in making the assumptions for a second trial.

62. Measuring on the drawing, Fig. 13, the total length available for the spring in the earliest cut-off position is found to be 5 feet 7 inches. Allowing 2 feet for attachments and idle coils, 3 feet 7 inches is available as the length of the active part of the spring at the earliest cut-off position. Subtracting the 11 inches (nearly) of elongation in this position leaves 2 feet 8 inches as the length available for the active part of the spring at zero load, which, divided by $\frac{5}{8}$ inch, gives about 50, showing that a suitable spring of $\frac{5}{8}$-inch wire must have about 50 coils.

63. Transposing the formula in Art. **61**,

$$n = \frac{s\,G\,d^4}{8\,W\,D^3} \qquad (1)$$

Inspection of this formula shows that the value of n varies inversely as the cube of D. That is, if D is made one-half as great, n will be eight times as large, all other conditions remaining the same. Conversely, D varies inversely as the cube root of n, so that if n is made eight times as great, D must be made one-half as great to avoid any change in results.

In the problem, it is desired to change the number of coils from 95 to 50, and the coil radius under these new conditions is to be found. Let n and D represent the number of coils and the coil diameter, respectively, before alteration, and n_1 and D_1 the corresponding values in the new spring. Then, according to the foregoing paragraph,

$$D_1 : D = \frac{1}{\sqrt[3]{n_1}} : \frac{1}{\sqrt[3]{n}};$$

or

$$D_1 = D\sqrt[3]{\frac{n}{n_1}} \qquad (2)$$

Now, $D = 2.5$, $n = 95$, and $n_1 = 50$. Substituting these values in formula **2**,

$$D_1 = 2.5\sqrt[3]{\frac{95}{50}} = 3.1 \text{ inches}$$

It will be found that a spring of $\frac{5}{8}$-inch wire of 3.1 inches mean diameter will require more space than is available on the wheel. So let D_1 be made still greater, say 3.5 inches.

64. By reference to the formula in Art. **60**, it is seen that the safe load varies inversely as the diameter of the coil. Hence, the safe load of the new spring is

$$W = 1{,}917.5 \times \frac{2.5}{3.5} = 1{,}370 \text{ pounds, nearly}$$

This being less than the greatest required load on the spring in the earliest cut-off position, the spring must be strengthened. As it is not desired to work the spring right up to its maximum safe load, it will be well to make it strong enough to carry 1,800 pounds. To do this, the diameter of the wire may be increased.

65. It is evident, from the formula in Art. **60**, that W varies directly as the cube of the diameter of the wire, or that the diameter of the wire varies as the cube root of the load, other conditions remaining the same. Hence, if the load is increased from 1,370 pounds to 1,800 pounds, the diameter of the wire must also be increased. If d and d_1 represent the old and the new diameters, and W and W_1 the old and the new loads, then

$$d_1 = d\sqrt[3]{\frac{W_1}{W}}$$

But, $d = \frac{5}{8}$, $W_1 = 1{,}800$, and $W = 1{,}370$. Substituting these values in the formula,

$$d_1 = \tfrac{5}{8}\sqrt[3]{\frac{1{,}800}{1{,}370}} = .684, \text{ or say } \tfrac{11}{16} \text{ inch}$$

Hence, the diameter of the wire must be increased to $\frac{11}{16}$ inch. Substituting this value, together with the other known values, in the formula in Art. **60**,

$$W = \frac{.3927 \times 50{,}000 \times (\frac{11}{16})^3}{3.5} = 1{,}823 \text{ pounds}$$

which shows that a spring of $\frac{11}{16}$-inch wire is amply strong. The deflection under a load of 1,823 pounds will be $\frac{1{,}823}{154.4} = 11.8$ inches. Substituting this value in formula **1**, Art. **63**, and solving for n, the number of coils,

$$n = \frac{11.8 \times 12{,}000{,}000 \times (\frac{11}{16})^4}{8 \times 1{,}823 \times (3.5)^3} = 51 \text{ coils}$$

Then the active part of the spring at zero load is $\frac{11}{16} \times 51 = 35\frac{1}{16}$ inches. Adding to this the elongation at the earliest cut-off position, found to be 10.415 inches, the total length of the active part of the spring at earliest cut-off is $35.063 + 10.415 = 45.5$ inches, nearly. The total length available for the spring at earliest cut-off being 5 feet $7\frac{1}{2}$ inches, there remains 1 foot 10 inches for attachments and idle coils, which should be sufficient.

As a satisfactory spring can be put in on the point of attachment assumed, the spring attachment will be retained as tried and as shown in Fig. 13. The outside diameter of the spring will be $3\frac{1}{2} + \frac{11}{16} = 4\frac{3}{16}$ inches, and the face of the governor wheel must be made wide enough between the sides of the arms and the edge of the rim to cover the spring.

66. The calculations thus far made have been the closest approximations possible from the data known, yet there is a chance that the final result may be inaccurate. Therefore, the governor must be built so that it may be readily adjusted to give the desired regulation when tested under normal working conditions.

The spring is generally secured at one end by a nut *e*, Fig. 14, formed with helical grooves on the outside, so that it may be screwed into one end of the spring *c* to a depth of several turns. Into this nut *e* is screwed a bolt *f* that passes through the rim of the governor wheel. The adjustment of the spring for the required regulation is accomplished mostly by means of the bolt *f*. If this bolt is screwed into the nut *e*, so as to increase the tension of the spring *c*, the regulation is made closer; that is, the difference between full-load speed and no-load speed is decreased, and at the same

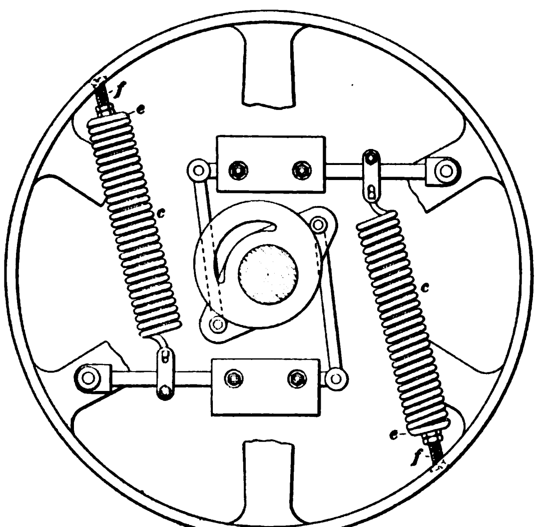


FIG. 14

time the mean speed of running is raised. Backing *f* out of *e*, so as to relieve the tension of *c*, makes the regulation less close and lowers the mean speed.

67. It is necessary, in addition to adjusting *f*, to be able to make some adjustment to get the desired mean speed without changing the closeness of the regulation. This may be accomplished in any one of the following ways:

1. By shifting the positions of the weights on the arms of the governor. Putting the weights closer to the pivots *o*,

Fig. 13, makes the engine run faster; while putting the weights farther from the pivots slows down the engine.

2. By changing the amount of the weights. This may be done in various ways. It is common to make the cast-iron weights hollow so that they may be filled with lead. The designed amount of the weight generally includes some lead, so that by cutting out lead or pouring more in, the amount of the weight may be adjusted. Increasing the weight slows down the engine, while decreasing it speeds up the engine.

3. By shifting the point of spring attachment along the arm of the governor. Shifting the spring in toward the pivot slows down the engine and at the same time makes the regulation a little closer, but not much. Shifting the point of spring attachment farther from the pivot speeds up the engine and at the same time makes the regulation a little less close.

4. By changing the number of active turns of the spring. This is done by screwing the nut *e*, Fig. 14, into or out of the spring *c*. Screwing *e* into the spring decreases the number of active turns and raises the speed of running without much effect on the regulation, what effect there is being to make it less close. Backing *e* out of the spring increases the number of active turns and lowers the speed of running, the effect on the regulation being to make it slightly closer.

A sufficient amount of adjustment of the finished governor should be allowed, in making the working drawings, to insure a final adjustment to suit the requirements. Adjustment of the bolt *f*, Fig. 14, should in every case be provided for. Of the adjustments for speed, 1 and 3 are often not provided for, 2 and 4 being depended on to give all the adjustment necessary.

68. When a governor is adjusted very closely to isochronism, it is apt to give trouble by being too easily disturbed by outside causes, such as the resistance of the valve, variation in the angular velocity of the shaft during each revolution, and so on. Governors have been built with an adjustment very close to isochronism, and frictional resistances introduced to steady them. Such a course is never to be recommended. While it may accomplish the purpose of steadying the governor against disturbances due to outside causes, its effect on the regulation of a nearly isochronous governor will be to cause the governor to hunt continually through a cycle or wave of speed, without settling down to a steady speed.

Dashpots give better results and furnish a proper last resort, if the means of adjustment just enumerated fail to accomplish the purpose; but the best plan is to use on the engine a balanced valve made as light as possible, provide ample lubrication, and in other ways keep down the friction of the valve, the governor, and the intervening mechanism to the lowest possible amount, and then provide a governor with plenty of power. To insure sufficient power, the moment of the centrifugal force of the weights must be made amply large and the governor adjusted to give a regulation not too close, in order that there may be sufficient power to overcome the resistance to its operation.

Stops should be provided on the governor wheel, against which the weights will rest at the desired limits of their movements.

69. It is common to build governor wheels so that they can be used with governors suited to either direction of rotation. If this is desired, the designer should determine the path of the eccentric center for the reversed rotation. This will give the necessary position of the eccentric pivot, and a boss may then be put on the wheel at this point. From the required movement of the eccentric, the movement of the governor can be laid out on the drawing board and the proper positions of the governor pivots, the points of attachment of the springs to the governor wheel, and of the stops can be found. Bosses should be provided on the wheel for the pivots and springs, if necessary. The stop for the direction of rotation that is not to be used will generally interfere with the motion of the governor for the desired direction of rotation.

If not, it may be put on. If it does interfere, bosses may be provided into which bolts may be screwed to serve as stops, if they should be required.

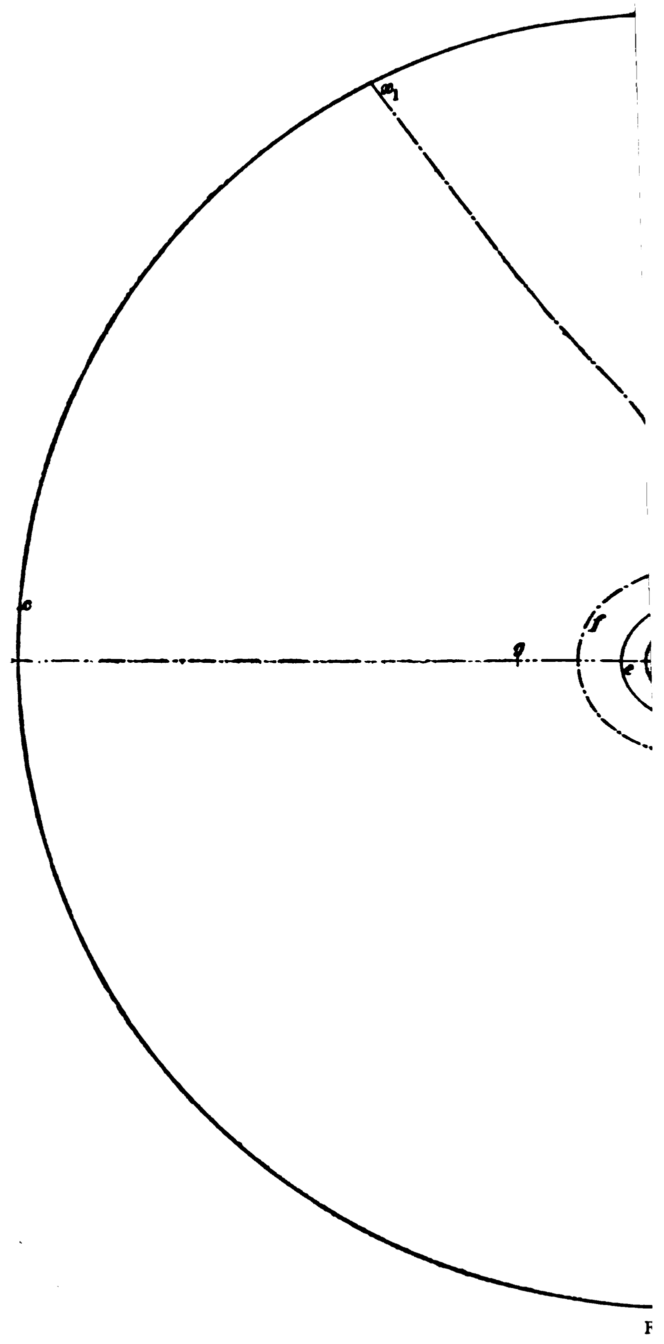
70. Design of the Inertia Governor.—The design of the inertia governor must also be worked out largely on the drawing board, and is, to a great extent, a trial process. The method will be illustrated by the design of a governor for an engine of 150 horsepower making 225 revolutions per minute.

First lay off on the drawing board the circle *c*, Fig. 15, representing the inner circumference of the rim of the governor wheel. The design of the governor is easier if the wheel is of large diameter; but as it is usually a single casting, the rim speed generally has to be kept down to the speed appropriate for flywheels of this construction, which is about 70 feet per second. As the wheel carries considerable additional load due to the governor, it is preferable to keep the rim speed below 70 feet per second, if possible.

71. Draw the circle *d*, Fig. 15, of the diameter of the shaft, and the circle *e* of the diameter of the hub of the governor wheel. It is usually desirable and often necessary that the spring should not draw across the hub. The outside radius of the spring is likely to be in the neighborhood of 2 inches. Therefore, draw a circle *f*, with a radius 2 inches greater than that of the hub circle *e*, about the center *o* of the shaft. Usually, the axis of the spring must be kept outside of this circle *f* at all positions of the governor.

It is most convenient to consider the crankpin, throughout the work on the design, as being on the dead center. The pin is shown at *g*, on the dead-center line *g o h*. From the layout of the valve diagram, it is found that the eccentric center for the latest point of cut-off is at *a*, and for the earliest point at *b*.

72. In order to make the governor work properly, the weight arm must be pivoted on the other side of the wheel from the crankpin. In order to avoid the friction of linkage, it is desirable that the eccentric should be mounted on the



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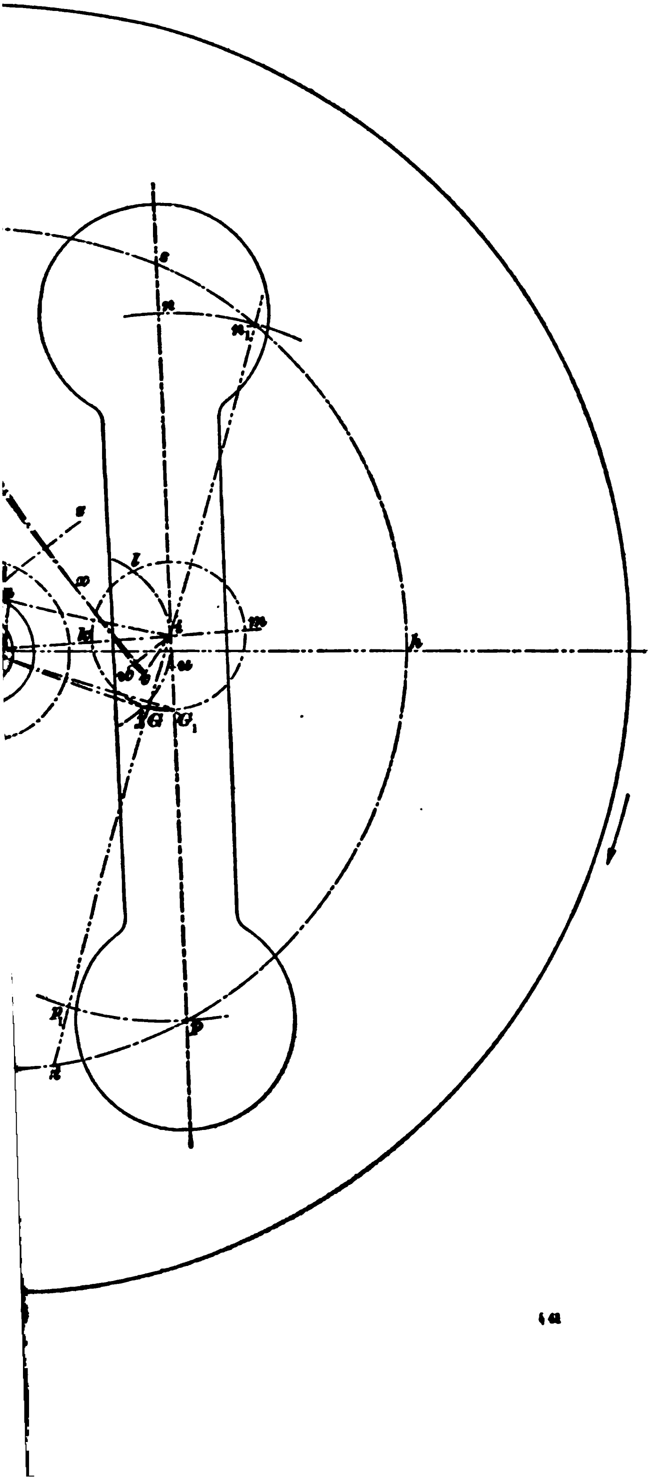
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governor weight, so that it may form practically one piece with it. The center of the eccentric pivot is, therefore, the center of the governor pivot also in the best forms of this type of governor. This necessitates the pivoting of the eccentric on the side of the shaft away from the crankpin. This fact must be borne in mind in the design of the valve gear if the use of the inertia governor is contemplated. The solution of the governor problem is facilitated by the use of a rather large radius for the eccentric path, which, in general, improves the operation of the eccentric also. In this case, the radius is taken as 9 inches.

Lay off on the drawing the path of the eccentric center *a b*. The radius of this arc being known, its center is at once located at *i*, which is the necessary center of pivot for both eccentric and governor.

73. The radius of the center of gravity of the governor about the center of its pivot must now be decided. As the governor is without gravity balance, it is desirable that this radius should be as small as possible in order to diminish the unbalanced moment of gravity about the pivot and thereby diminish the undesirable periodic oscillations of the governor produced by this moment. On the other hand, if the center of gravity were at the center of the pivot, the centrifugal force would have a zero moment and the governor would have no power. The farther the center of gravity is from the center of the pivot, the greater is the power of the governor for a given weight; this is desirable. As a fair compromise, a radius of 4 inches for the center of gravity is adopted in this case; this is about as large a radius as should be used.

The distance of the center of gravity from the center of the pivot having been decided, the circle *j k l m* is drawn about *i* as a center with this distance as a radius. Then the center of gravity of the weight must lie on the circle *j k l m*.

74. In order that the inertia may act with the centrifugal force, as explained in Art. **45**, the tangent to the path of the center of gravity of the weight, at the center of gravity, must pass between the center of the shaft and the center of the pivot when the weight follows the pivot; when the weight precedes the pivot, the tangent must not pass between the pivot and the center of the shaft.

To secure this condition, draw the straight line $o\,i\,m$, and on $o\,i$ as a diameter draw the arc $l\,i\,j$. Then $o\,i\,m$ intersects $j\,k\,l\,m$ in m and k, and $l\,i\,j$ intersects $j\,k\,l\,m$ in l and i. To fulfil the requirements, the center of gravity of the governor weight must lie above k and below l, or below m and to the right of j. The eccentric path and crankpin being as shown, the direction of rotation is necessarily that of the arrow. In this case, the center of gravity is located on the arc $m\,j$, that is, preceding the pivot i in the direction of rotation.

75. This location of the center of gravity being decided on, it appears that its path must lie close to j, because if the path lies at all up toward m, the arm of the centrifugal force which acts in a radial direction, and about i, will decrease so much as the governor swings from the late to the early cut-off position that it will be impossible to make the centrifugal force moment increase from late to early cut-off, as is necessary.

For a trial, the center of gravity of the weight arm is therefore located at G for the late cut-off position. This also locates G_1, the early cut-off position of the center of gravity of the weight arm. For, the center of gravity swings in a circle of radius $i\,G$, in the direction from G to G_1, from late to early cut-off and the angle $G\,i\,G_1$ must therefore equal the angle $a\,i\,b$.

76. The next step is to locate trial positions of the axis of the weight arm for both late and early cut-offs. As the choice of these positions depends on the location of the spring, the method of locating the spring must now be considered.

The following is applicable to both centrifugal and inertia governors. Let a, Fig. 16 (a) and (b), be the center of the shaft, b the latest cut-off, and c the earliest cut-off position, of the center of gravity of the weight arm. In Fig. 16 (a) is shown the form assumed by the balanced shaft governor, and in Fig. 16 (*b*) the form assumed by the inertia governor. Let *d* be the center of the weight-arm pivot. Then *b c* is a circular arc with its center at *d*. Locate *e*, a suitable point of attachment of the spring to the weight arm, in the early cut-off position. Draw *e f* so that the angle *d e f* equals the angle *d c a*. In general, *e* must be so located that, if the spring axis lies in *e f*, there will be plenty of room for the spring between *e f* and the hub of the wheel; and, if possible, some margin also for adjustment of the point of spring attachment.

77. From early to late cut-off, the point of spring attachment swings about *d* as a center, in the direction from *c* to *b*,

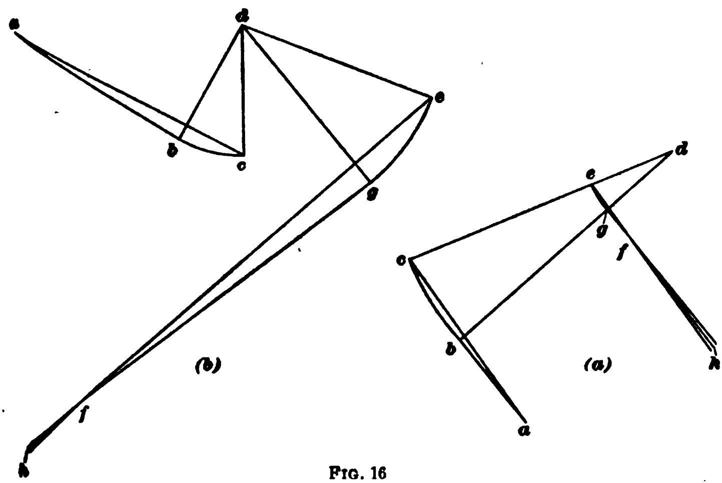

FIG. 16

in a circular arc of radius *d e*. The point *g* is therefore the late cut-off position of the point of spring attachment, the angle *g d e* being the same as the angle *b d c*. Through *g* draw *g f* so that the angle *d g f* equals the angle *d b a*; *g f* and *e f* intersect at *f*. Then, *f* is the theoretically correct point of attachment of the spring to the governor wheel. With practical dimensions *f* generally comes too close to the governor to allow the spring to be attached there; but *e* should be so chosen that the angle *e f g* is very small. Then, any convenient point in *f h*, the bisector of the angle *e f g*, is a close enough approximation for all practical purposes.

78. By trial, a position of the axis of the weight arm in the early cut-off position must now be located on the drawing board. This axis may be taken as passing through G_1, Fig. 15, that being the most convenient construction. It must also lie in such a position that a point for the attachment of the spring to the weight arm may be chosen not much farther from i than G_1, and when the location of the spring axis from this point is determined by the method of Fig. 16, the spring axis must not go inside of the circle f, Fig. 15. Also, the more room there is across the governor for the insertion of the spring, the better.

In the example, the early cut-off position of the axis of the governor is located at $n\,p$. In this case, the axis passes through i, the pivot of the governor. This is convenient, but not essential. Since the weight arm and eccentric swing together, the axis for late cut-off, $n_1 p_1$, is drawn at an angle to $n\,p$ equal to the angle $a\,i\,b$.

79. The cylindrical ends of the governor weight are assumed to be 12 inches in diameter. This dimension may often be made less. The radius of the end cylinders being determined, this distance, together with some additional allowance, should be subtracted from the radius of the inner circumference of the wheel rim, and an arc of a circle $q\,h\,r$, with the resulting dimension as a radius, should be struck about o as a center. This arc intersects $n\,p$ in s and p; and $n_1 p_1$ in n_1 and t. As n_1 is nearer than t to i, it is taken as the center of the round end of the weight, in the late cut-off position; and $n\,n_1$ is the circular arc in which the center of the weight moves about i as a center. The position of the center of the round end of the weight in the early cut-off position is n. Similarly p_1 and p are the positions of the center of the other cylindrical end of the governor weight, at late and early cut-off positions, respectively.

80. The outline of the governor may now be drawn in the early cut-off position, as shown in Fig. 15. The middle body of the weight may be taken as a cast-iron bar of rectangular cross-section. In the example, the bar is taken as

6 inches wide and $1\frac{5}{16}$ inches thick. Measure the distance $o\,G$, which, in the example, is $9\frac{11}{16}$ inches = .8073 foot.

Let W be the total weight of weight arm, in pounds. Then, by the formula for centrifugal force,

$$F = .00034\,W \times (225)^2 \times .8073 = 13.90\,W \text{ pounds}$$

In the late cut-off position, 10,000 inch-pounds is assumed to be a large enough centrifugal force moment in this case; and as greater power in the governor would require greater strength in all its parts and attachments, the governor is designed for only the necessary 10,000 inch-pounds.

Draw the straight line $o\,G$ and measure the perpendicular distance from i to this line, which in this case is 4 inches; this distance is the arm of the centrifugal force about i. Then, $13.90\,W \times 4 = 10{,}000$, whence $W = 179.9$ pounds.

The weight of the arm connecting the weights is found to be about 25 pounds per linear foot. The weight of all bosses and the eccentric may, with sufficient accuracy, be neglected.

81. The mean length of the bar connecting the weights may, with sufficient accuracy, be taken as the average of its length between round ends on its axis and of its length between round ends along its outside edge. In the example, the mean length of the bar is thus found to be about 2 feet $3\frac{3}{8}$ inches = 2.281 feet. Consequently, the weight of the bar is 57 pounds.

Let w = weight of round end that is centered at n;

w_1 = weight of round end that is centered at p.

Then, $$w + w_1 + 57 = 179.9$$

The center of gravity of the bar is at the middle of its figure, which is at u, Fig. 15. Consequently, the static moment of the bar (that is, the moment when the governor is at rest) about G_1 is

$$57 \times G_1 u = 57 \times 2.375 \text{ inches (by measurement)}$$
$$= 135.375 \text{ inch-pounds}$$

Similarly, the static moments of the ends about G_1 are

$$w \times G_1 n = 21.5\,w \text{ (by measurement)}$$

and $$w_1 \times G_1 p = 17\,w_1 \text{ (by measurement)}$$

82. In order to bring the center of gravity of the weight as a whole to G_1, the algebraic sum of the static moments of all parts of the weight about G_1 must equal zero. Writing this in the form of an equation,

$$17\,w_1 - 21.5\,w - 135.375 = 0$$

But, $$w + w_1 + 57 = 179.9$$

Solving these two equations for w and w_1, $w = 50.75$ pounds, and $w_1 = 72.15$ pounds.

Let t = thickness of w, in inches;

t_1 = thickness of w_1, in inches.

Then, $$t \times \frac{\pi \times 12^2}{4} \times \frac{450}{1{,}728} = 50.75;$$

whence $t = 1.72$ inches. Similarly, $t_1 = 2.45$ inches.

83. A suitable point for the attachment of the spring to the weight arm, in the early cut-off position, must now be found by trial on the drawing board. This point must fulfil the following conditions:

1. It must, generally, not be very far from i in order that a satisfactory spring may be designed.
2. The tension on the spring must increase as the governor weight swings from the late to the early cut-off position.
3. The line of the axis of the spring, as located by the method of Arts. **76** and **77**, must not go inside the clearance circle f, drawn about the hub in Fig. 15.
4. Along the line of the spring axis, there must be plenty of room between the governor weight and the rim of the wheel for the spring and attachments.

The point v, Fig. 15, is selected as the position of the point of spring attachment in the early cut-off position. As the eccentric center swings from b to a, that is, from early to late cut-off, the point of spring attachment swings in the same direction from v to w, $v\,w$ being a circular arc about i as a center, and the angle $v\,i\,w$ being equal to the angle $a\,i\,b$.

84. By the method of Fig. 16 (b), the theoretically correct point of attachment of the spring to the wheel is now found to be at x, Fig. 15. This point will have to be located instead somewhere on the line $x x_1$, the bisector of the angle $v x w$.

For the purpose of determining the moment of the spring about the center i, the line $x w$ may be used as a close enough approximation to the true line of the spring axis in the late cut-off position. By measuring the perpendicular distance from i to $x w$, the arm of the spring pull at late cut-off is obtained. This distance is 2.5 inches.

As in the case of the centrifugal governor, this governor is designed for perfect isochronism, with provision for adjustment under actual running conditions to secure as much stability as may be found necessary. The spring pull moment must therefore be designed to balance the centrifugal force moment at 225 revolutions per minute at all positions of the governor.

85. Since the moment F_l of the centrifugal force in the late cut-off position was designed to be 10,000 inch-pounds, the tension on the spring in the same position is given by

$$T_l = 10{,}000 \div 2.5 = 4{,}000 \text{ pounds}$$

Draw the straight line $o\,G_1$, which is by measurement $10\frac{7}{8}$ inches = .906 foot; this is, then, the radius of the center of gravity of the weight about o, in the early cut-off position. Then the centrifugal force in the early cut-off position is

$$F_e = .00034 \times 179.9 \times (225)^2 \times .906 = 2{,}805 \text{ pounds}$$

Measure the perpendicular distance from i to $o\,G_1$. This is the arm of the centrifugal force about i in the early cut-off position, and, in the example, is $3\frac{13}{16}$ inches. Then the moment of centrifugal force at early cut-off is

$$M_e = 2{,}805 \times 3\tfrac{13}{16} = 10{,}694 \text{ inch-pounds, nearly}$$

86. Measure the perpendicular distance from i to $x v$ which will give nearly enough the arm of the spring pull at early cut-off, and which, in the example, is $2\frac{5}{16}$ inches. Then the tension on the spring in the early cut-off position is

$$T_e = 10{,}694 \div 2.3125 = 4{,}624 \text{ pounds, nearly}$$

The stretch of the spring from the late to the early cut-off position is approximately $w v$, which measures $\frac{3}{4}$ inch and is nearly enough correct. Then the load of the spring per inch of stretch is

$$\frac{4{,}624 - 4{,}000}{.75} = 832 \text{ pounds per inch}$$

If a spring is designed to meet this condition when attached to the assumed point on the weight arm and to a point on the wheel in the line $x x_1$, and is set up, to an initial tension, at the early cut-off position of 4,000 pounds, the centrifugal force moment and the spring pull moments will not only balance at 225 revolutions per minute, at the extreme positions of the governor for which they are calculated, but they will balance at that speed for all positions of the governor and fulfil the conditions of isochronism.

87. It is now necessary to see whether a spring to fulfil the above requirements can be so designed that it will go in the space available for it on the governor wheel. As it may at some time be desirable to increase the tension on the spring considerably in order to govern the engine isochronously at some higher speed than originally intended, the spring may be designed for a maximum load of 7,000 pounds, instead of 4,626 pounds, as required for the given speed. The method to be followed in designing this spring is the same as that given in Arts. **57** to **66.**

88. Where the spring may be attached to a point well inside the rim of the wheel, it is well to lay out the wheel so as to bring a spoke at this point, and attach the spring to a boss on the spoke, rather than to carry it to the rim by a tie-rod. The reason for this is that the centrifugal force of the mass of the spring itself causes the axis of the spring to bow out from the center of the shaft when the engine is running. This puts a tension on the spring, which tends to disturb its action. The greater the unsupported length of the spring and attachments the greater is this disturbing effect.

The eccentric in the example is simply a pin mounted on an arm projecting from the body of the governor weight. The remarks in the last paragraph of Art. **67,** on the adjustment of the balanced shaft governor, apply also to the inertia governor.

89. In order to be able to change the position of the center of gravity of the weight, the cylindrical ends may be made as shown in Fig. 17, the most of the weight of each end being in the form of removable disks. The disks are often made of lead.

If it is desired to shift the center of gravity toward one end, disks may be put into that end, or if it is desired to keep the total weight constant, the weights may be taken out of one end and put into the other.

The total weight may be increased or decreased, without changing the position of the center of gravity, by adding or subtracting weights at both ends in reverse ratio to the distances of their centers from the center of gravity of the whole weight.

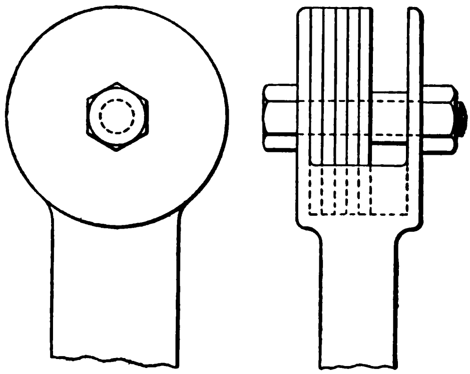

FIG. 17

STEAM-ENGINE DESIGN

(PART 1)

DATA AND CALCULATIONS

PRELIMINARY DATA

1. The designer of an engine has for his preliminary data: (1) The class of service for which the engine is intended, whether it is for a special case, as marine, mill, electrical, locomotive, hoisting, etc., or whether it is to be put on the market and sold wherever possible, to cover a large variety of classes; (2) the rated indicated horsepower; (3) the necessary economy; (4) the allowable fluctuation of speed; (5) occasionally, the boiler pressure; and (6) the type of engine desired—that is, whether simple or compound, horizontal or vertical, high- or low-speed, Corliss or slide-valve gear, etc.—which depends to a considerable extent on the nature of the preceding data.

For the economical range of load, the designer must determine: (1) The boiler pressure, if not already known; (2) the back pressure; (3) the point of cut-off at rated indicated horsepower; (4) the approximate piston speed; (5) the clearance, and (6) the amount of compression to be employed.

In order to design a simple engine after having obtained the foregoing data, it will be necessary to draw a theoretical indicator diagram and determine the mean effective pressure; then, the proportions of the cylinder can be calculated, and the design of the other parts will readily follow. For a compound or triple-expansion engine, a more complicated process is necessary.

2. The Boiler Pressures for Different Types of Engines.—The **boiler pressure** for an engine to be designed may be fixed beforehand, if it is known that the engine is to have steam furnished by an existing boiler or set of boilers carrying a definite pressure. In case the boiler pressure is not known, then experience has fairly definitely determined, for each type of engine, a range of pressure outside of which it is not desirable to go. This range is about as follows:

Type of Engine	Gauge Pressure, in Pounds
Simple	70 to 120
Compound	100 to 150
Triple-expansion	150 to 200 or higher
Quadruple-expansion	200 or higher
Locomotive	160 to 210

The ranges just given represent the best practical results for all conditions under which a steam engine is to operate, the chief consideration usually being the number of cylinders in series in which the expansion of the steam is to take place. As to choice of pressure within any of the ranges indicated, a low steam pressure is desirable, if simplicity and low first cost are the prime considerations; but if economy in weight and space are needed, a high steam pressure is necessary. The best all-around economy in fuel and other running expenses for any particular type of engine will usually be obtained with a steam pressure about the middle of the range for that type of engine.

The initial pressure in the engine cylinder will be less than the boiler pressure, on account of the loss caused by resistance to flow through the steam pipe and connections. Ordinarily, the loss may be taken at about 8 per cent. of the boiler pressure.

Even if the engine is rated on the basis of a lower pressure, the parts should be designed to carry safely a pressure of at least 100 pounds per square inch, gauge, with a back pressure as low as that ordinarily attained by a condenser, say 2 pounds per square inch, absolute; that is, with an unbalanced pressure of 112.7, say 113, pounds per square inch, for the engine may at some time be run with a condenser, and, when starting, stopping, or running at low speed, the unbalanced pressure may become equal to the full boiler pressure plus the condenser pressure.

3. Piston Speed.—The best practical results may be attained by using the following **piston speeds:**

Type of Engine	Piston Speed, in Feet per Minute
Stationary, small	300 to 600
Stationary, medium size	600
Stationary, large	750
Marine	850 to 900
Locomotive	600 to 1,200

4. Economical Ratio of Expansion.—In a simple engine, the greatest economy of steam occurs with a **ratio of expansion** varying from about 3 to 5, and within this range of the ratio of expansion, the *steam economy* does not vary much. It is customary to rate an engine at its most economical load, which, in the case of a simple engine, will occur with a ratio of expansion between 3 and 5, except with the plain slide-valve engine having no cut-off mechanism. As it is more common to run a steam engine underloaded than overloaded, most of the medium- and high-speed automatic cut-off engines constructed at the present time by American builders are rated on a ratio of expansion of about 3, with the expectation that the engine will usually run inside of its economical range. It should be remembered that the ratio of expansion mentioned here is the true ratio of expansion; and this, at a value of 3, with the large clearance used in high-speed automatic cut-off engines, brings the cut-off at about .25 stroke at rated load.

As the latest cut-off of single-wristplate Corliss engines is between .4 and .5 stroke, the makers of such engines usually rate them on cut-off at .2 stroke, in order to give them some overload capacity. This makes the true ratio of expansion of such engines at rated load between 4 and 5, with the usual clearance.

5. For a compound engine, the total combined theoretical indicator diagram may be drawn, and this may be divided between the cylinders. The division of the diagram shows the proper ratios of expansion for each cylinder. If the maximum of steam economy is desired, the diagram should be divided so as to give about an equal range of temperature to each cylinder. If the greatest uniformity of rotative speed is desired, the diagram should be divided so as to give about the same amount of work to each cylinder. If both are desirable, an intermediate division of the diagram becomes necessary. A simple extension of the method gives the ratios of expansion for triple- and quadruple-expansion engines.

6. Clearance.—The term *clearance* in connection with the steam engine is used in two senses. The clearance may be the distance between the piston and the cylinder head when the piston is at the end of its stroke, or it may represent the volume between the piston and the valve when the engine is on dead center. To avoid confusion, the former is called **piston clearance,** and the latter is simply termed **clearance.** Piston clearance is always a measurement expressed in parts of an inch. Clearance, however, is a volume, and it is usually taken as a percentage of the volume swept through by the piston per stroke. Thus, a clearance of 7.5 per cent. signifies that the volume just defined is 7.5 per cent. of the volume swept through by the piston during one stroke. The usual values of the clearance in various types of engines as found in practice are:

Type of Engine	Clearance Per Cent.
Four-valve, experimental	1
Corliss and other drop cut-off	1.5 to 3.5
Medium-speed	3 to 8
High-speed, with long slide valves	4 to 12
High-speed, with short, plain slide valves	7 to 15

Within the ranges just stated, the large engines of each type will have the smaller clearances, and the small engines will have the larger clearances. High rotative speed tends to large clearance, and low rotative speed to small clearance. In slide-valve engines, piston valves give larger clearances than flat valves.

It is usually desirable to reduce the clearance to a minimum. However, with the early exhaust closure of single-valve automatic engines, large clearance reduces the pressure in the cylinder at the end of compression below what might otherwise be objectionable; and, with high-speed engines, the large clearance also reduces the danger due to water in the cylinder.

7. When used as a linear distance, clearance is usually spoken of as *piston clearance* or, sometimes, *mechanical clearance;* it is the shortest distance in the direction of the stroke between the piston when at the end of the stroke and the nearest cylinder head, and should be made as small as possible. On small stationary engines, this distance may be $\frac{1}{4}$ inch, and it rarely exceeds $\frac{1}{2}$ inch on the largest marine engines. In some cases in actual practice, with low-pressure cylinders 7 feet in diameter and conical pistons, this clearance is only $\frac{3}{8}$ inch.

For the purpose of design the piston clearance may generally be found by the formula:

$$c = a + b\,x,$$

in which c = piston clearance, in inches;

x = number of bearings, or joints, between piston and crank-shaft, where there can be play;

a = $\frac{1}{8}$ inch for engines of 35 horsepower or less; $\frac{3}{16}$ inch for engines between 35 and 100 horsepower; and $\frac{1}{4}$ inch for engines above 100 horsepower;

b = $\frac{1}{32}$ inch for engines of 35 horsepower, or less; $\frac{3}{64}$ inch for engines between 35 and 175 horsepower; and $\frac{1}{16}$ inch for engines above 175 horsepower.

The horsepowers just stated are taken at rated load; that is, the load for which the engine was designed. For very fast-running engines the allowance for each bearing may be doubled.

ENGINE CALCULATIONS

BACK PRESSURE AND POINT OF EXHAUST CLOSURE

8. Back Pressure and Compression.—In a well-designed non-condensing engine, the **back pressure** should not exceed 16 or 17 pounds per square inch, absolute. For a condensing engine, the back pressure may be from 2 to 4 pounds per square inch, absolute.

The method of finding the proper amount of compression depends on the class of valve gear used, since the point of exhaust closure may remain the same at all loads, as in the Corliss or other independent cut-off valve gear; or, the exhaust closure may change with the load, as in the single swinging-eccentric gear controlled by a shaft governor.

The case in which the point of exhaust closure does not change will first be considered. Taking the ratio of expansion at rated load appropriate to the case, as explained in Art. 4, and the expansion curve as $p v$ = a constant, then the pressure at the end of the stroke will be the initial absolute steam pressure in the cylinder divided by the ratio of expansion.

The next step is to find the weight of the piston, piston rod, and crosshead. This weight may be estimated with all necessary accuracy by examining the records of weights of these parts in engines of similar type and, as near as possible, the same power. A designer usually has access to such data in the office records of his employer.

Let w = weight of reciprocating parts, that is, sum of weights of piston, piston rod, crosshead, and half the connecting-rod, in pounds;
N = number of revolutions per minute;
r = length of the crank, in feet;
F = total inertia pressure at end of stroke, in pounds.

The reciprocating parts, on account of their inertia at the end of the stroke, will exert a pressure in the direction of motion. This inertia pressure may be expressed in pounds with sufficient accuracy by the following formula, which is based on the assumption that the connecting-rod is of infinite length and is explained in *Mechanics of the Steam Engine*.

$$F = .00034\, w\, N^2\, r \qquad (1)$$

Let A = piston area, in square inches, taking an approximate value from the records, as just explained;

F_1 = inertia pressure at end of stroke, in pounds per square inch of piston area.

Then, formula **1** becomes

$$F_1 = \frac{.00034\, w\, N^2\, r}{A} \qquad (2)$$

Let p_2 = absolute forward steam pressure at the end of the stroke, in pounds per square inch; that is, the initial pressure divided by the ratio of expansion;

p' = total forward absolute pressure on the piston at the end of the stroke, in pounds per square inch.

Then,
$$p' = p_2 + F_1 \qquad (3)$$

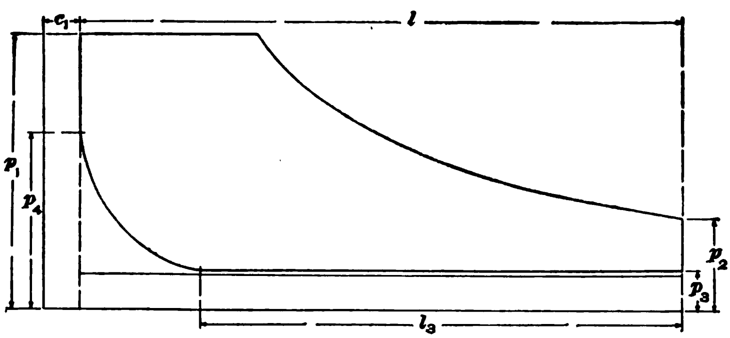

FIG. 1

9. Point of Exhaust Closure.—In Fig. 1 is shown a diagram that indicates the pressures and percentages of stroke. The initial pressure is p_1, the pressure at the end of expansion p_2, the back pressure p_3, and the compression pressure p_4, all absolute pressures in pounds per square inch. The length of the stroke is l, the part of the stroke from the beginning of exhaust to the point where the exhaust valve closes is l_3, and the clearance is c_1. As l_3 and c_1 are usually expressed as percentages of the stroke, then l, or 100 per cent. of the stroke, is expressed as 1.

To cushion the reciprocating parts properly, the pressure at the end of the stroke due to compression should equal p'. Remembering that the volumes in the cylinder are proportional to the percentages of the lengths of stroke, the pressure p' times the clearance c_1 equals the absolute back pressure p_3 at exhaust closure multiplied by the clearance plus the portion of the stroke yet to be completed, which is $(1 + c_1 - l_3)$, or, expressed as a formula, $p'c_1 = p_3(1 + c_1 - l_3)$, which reduces to

$$l_3 = 1 - \frac{(p' - p_3)c_1}{p_3} \qquad (1)$$

Lacking the practical data that would have to be known to use the formula just given, a trial value of the pressure in the cylinder at the end of compression may be assigned by the use of the following empirical formula:

$$p_4 = (.0025N + .2)p_1 \qquad (2)$$

in which p_1 = initial steam pressure in the cylinder, in pounds per square inch, absolute;

p_4 = absolute steam pressure in the cylinder at end of compression;

N = number of revolutions per minute.

Then, as p_4 should equal p', this value may be used in formula **1**, giving

$$l_3 = 1 - \frac{(p_4 - p_3)c_1}{p_3} \qquad (3)$$

10. After the design of an engine has proceeded far enough, the calculation based on the weight of reciprocating parts can be made from the results obtained. If $p' = p_4$, that is, if the total forward pressure on the piston at the end of the stroke equals, approximately, the back pressure due to compression at the end of the stroke, the result will be sufficiently accurate for present use; but if these two quantities are very much different, such modifications should be made as may be necessary to bring them approximately equal, provided the engine is one of high or medium speed.

With low-speed engines, such as the Corliss, it is usually unnecessary to pay much attention to the equality of the steam pressure at the end of compression, with the total forward pressure on the piston. In these, if the exhaust closure takes place a little before the admission of live steam at the same end, say at 95 per cent. of the stroke, the results will be satisfactory for all practical purposes.

11. Exhaust Closure in Engine With Single Swinging Eccentric.—With a single swinging eccentric a different method must be followed. It may be assumed that the power will not be sufficient to run the engine against its own friction with a cut-off earlier than $\frac{1}{20}$ stroke, and that the engine will therefore never run with a cut-off earlier than this. Then, at this earliest cut-off, the back pressure at the end of the stroke due to compression should not exceed the initial steam pressure in the cylinder; otherwise, the valve may be forced from its seat. To avoid this, the following formula, in which the values of the different quantities are the same as in Art. **9**, should be used:

$$l_3 = 1 - \frac{(p_1 - p_3)c_1}{p_3} \qquad (1)$$

This formula gives the percentage of the stroke at which exhaust closure should occur at the earliest cut-off. Then, by means of the valve diagram, the resulting point of exhaust closure may be found for the rated load and ratio of expansion. This will be illustrated by an example in the succeeding article.

When only a rough approximation is required, the following formula is sometimes used:

$$p_4 = \frac{p_1 + 16}{2} \qquad (2)$$

FUNDAMENTAL ENGINE CALCULATIONS

12. Calculations for Simple Non-Condensing Engine.—The general method of procedure in a design will be illustrated in the following example:

Let it be required to determine the diameter of cylinder, length of stroke, and number of revolutions of a simple, moderate-speed, non-condensing engine, having independent cut-off, slide-valve gear, and fixed compression, to develop 100 indicated horsepower at the rated load.

As there is no particular reason in this case for going to an extreme in steam pressure, a pressure in about the middle of the range for a simple engine will be taken; namely, 100 pounds gauge pressure or 114.7 pounds, absolute. Then, assuming a ratio of expansion of 3, the terminal pressure in the cylinder will be $114.7 \div 3 = 38.2$ pounds, absolute.

Suppose that the designer has access to the records and drawings of two engines, similar to the one in hand, on which appear the following data:

Rated Horsepower	Weight of Reciprocating Parts	Area of Piston	Length of Stroke
	Pounds	Square Inches	Inches
60	450	78.5	18
130	542.8	159.48	22

It will be sufficiently accurate for the purpose to assume that the weight of the reciprocating parts of the proposed engine may be found by interpolating between the weights of the two engines just given. This may be done by comparing the difference in horsepower with the difference in weight. The difference in the horsepowers of the known engines is $130 - 60 = 70$, and that in the weights is $542.8 - 450 = 92.8$. Hence, for a difference of 70 horsepower, there is a difference of 92.8 pounds in the weights of the reciprocating parts. The difference between 100 horsepower and 130 horsepower is 30 horsepower. The difference in weight corresponding to this difference in horsepower is $\frac{30}{70} \times 92.8$, or 39.8 pounds. This gives the weight of the reciprocating parts for an engine of 100 horsepower as 542.8 − 39.8 = 503 pounds.

Similarly, the area of the piston will be the larger area less $\frac{30}{70}$ of the difference between the given areas, or 159.48 − $\frac{30}{70}$ (159.48 − 78.5) = 124.77, say 124.8, square inches.

13. In the course of a mechanical design, it often becomes necessary to estimate or assume approximate values before final values can be calculated. This can be done most satisfactorily by the foregoing process of interpolation from existing designs. Such interpolation does not usually enable the designer to determine values with enough accuracy to use them as working dimensions, but it is of great service in enabling him to reach a working approximation to values that enter into the calculations. In the absence of information from previous designs, necessary preliminary estimates must be made by unaided judgment; then, designing becomes largely a trial process, in which the calculations must be made and remade until the final result is satisfactory.

By interpolating again, the approximate length of stroke of the proposed engine is found to be $22 - \frac{30}{70}(22 - 18) = 20\frac{2}{7}$ inches. As this is an inconvenient dimension, 20 inches may be taken. Then, r, the length of the crank in feet, is $10 \div 12$. For a piston speed of 600 feet per minute, from Art. **3**,

$$N = \frac{600 \times 12}{20 \times 2} = 180 \text{ revolutions per minute}$$

From the values thus found, the pressure of the reciprocating parts at the end of the stroke may be found by means of formula **1**, Art. **8**; thus, $F = .00034 \times 503 \times (180)^2 \times \frac{10}{12} = 4{,}617.54$ pounds, and the corresponding pressure per square inch of piston is $4{,}617.54 \div 124.8 = 37$ pounds. The total forward pressure per square inch of piston, at the end of the stroke, is then $38.2 + 37 = 75.2$ pounds.

14. It is now necessary to assign a value to the clearance. Referring to Art. **6**, the small size of the engine as well as the form of valve gear would indicate that the clearance must be large, while the moderate rotative speed would tend to give small clearance. Hence, a value well up in the range for medium-speed engines, but not at the very top, may be assigned to the clearance. Let a clearance of 6 per cent. be assumed, and, from Art. 8, a back pressure of 17 pounds, absolute. Then, the necessary point of exhaust closure is at the fraction of the stroke given by formula 1, Art. 9; thus,

$$l_s = 1 - \frac{.06\,(75.2 - 17)}{17} = .795, \text{ or } .8, \text{ nearly}$$

15. There is now sufficient data to draw the theoretical diagram, as shown in Fig. 2, the scale to which the diagram

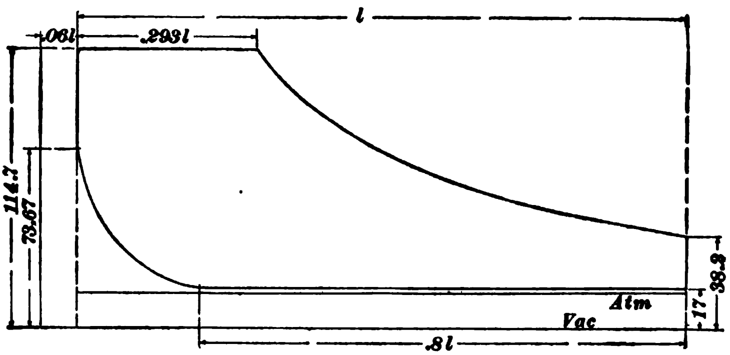

FIG. 2

should be drawn being entirely a matter of convenience. Let the pressure scale in this case be 50 pounds to the inch and the volume scale such that the diagram is 5 inches long; then, the clearance is $5 \times .06 = .3$ inch.

The diagram may now be drawn, assuming that the expansion and compression curves follow the law pv = a constant. By either measurement or calculation, the *mean effective pressure*, or the M. E. P., of the diagram, Fig. 2, is found to be 54 pounds per square inch. On account of the cylinder condensation and other losses, the M. E. P. given by the theoretical card is never attained by the actual engine. To find the probable M. E. P. of the actual engine, the M. E. P. of the theoretical card must be multiplied by a factor, the magnitude of which depends on the type of engine.

For jacketed engines, with Corliss valve gear or other quick-acting drop cut-off gears, or multiple-ported valves with good-sized ports for swinging eccentric engines, a factor of .94 is used.

For jacketed engines, with slide valves, either single-ported with large travel, large ports, and late cut-off, or double-ported valves in which the port opening in the early part of the stroke is not particularly good, a factor of .9 to .92 is used.

For unjacketed engines, with plain D-slide valve for cut-off about one-half stroke or earlier, or where the port opening is rather small in order to secure small travel with an unbalanced valve, a factor of .8 to .85 is used.

The foregoing values of the diagram factor are taken from Seaton, and represent conservative practice. Many American designers use factors somewhat higher than these, and employ the same factor for both jacketed and unjacketed engines of any particular type. This is done because it is supposed that the jacket affects the quantity of steam taken into the cylinder, rather than the area of the indicator diagram.

In the case of the simple engine of 100 horsepower under consideration, the factor will be about .94; then, the probable M. E. P. is $54 \times .94 = 50.76$ pounds per square inch.

16. The formula for the indicated horsepower of an engine is

$$\text{I. H. P.} = \frac{PLAN}{33{,}000} \qquad (1)$$

in which I. H. P. = indicated horsepower;

P = mean effective pressure, in pounds per square inch;

L = length of stroke, in feet;

A = area of piston, in square inches;

N = number of revolutions per minute.

The area of the piston or cylinder may be found by transforming formula **1**; thus,

$$A = \frac{33{,}000 \times \text{I. H. P.}}{PLN} \qquad (2)$$

As the product of the length of the stroke L and the number of revolutions N equals the piston speed in feet per minute, the piston speed may be substituted for $L N$ in the formula. Then, in the case of an engine of 100 horsepower and a piston speed of 600 feet per minute, formula **2** gives

$$A = \frac{33{,}000 \times 100}{50.76 \times 600} = 108.4 \text{ square inches}$$

The area A just found is sufficiently accurate for practical purposes as a preliminary estimate; however, it must be taken as the average area of the piston. As the effective area of one side is diminished by the area of cross-section of the piston rod, the area of the other, or free, side of the piston must be increased accordingly.

It will be accurate enough for the purpose to make this correction in the following manner:

Let D = diameter of the cylinder, in inches;

A = area of the cylinder, in inches.

Then, $A = .7854\, D^2$, or $D = \sqrt{A \div .7854}$

Hence, in this case,

$$D = \sqrt{108.4 \div .7854} = 11.75 \text{ inches,}$$

which is near enough for the present purpose.

Let d = diameter of the piston rod, in inches;

p = maximum unbalanced pressure on the piston, in pounds per square inch.

Then,

$$p = 114.7 - 2 = 112.7, \text{ say } 113, \text{ pounds}$$

The diameter of the piston rod may be found approximately by the formula

$$d = .02\, D \sqrt{p}$$

from which

$$d = .02 \times 11.75 \times \sqrt{113} = 2.5 \text{ inches}$$

The area of a circle of this diameter is 4 9 square inches. Let A be the area of the free side of the piston; then,

$$A = 108.4 + \frac{4.9}{2} = 110.85 \text{ square inches,}$$

from which, the required diameter of the piston and cylinder is 11.89 inches.

If the design is one of a line of engines to be built in large numbers and in standard sizes, the diameter of the cylinder would probably be taken as 12 inches. The effect of this change to even dimensions will be that, at the rated load, the ratio of expansion will differ somewhat from 3; but there is no objection to this slight departure from the ratio of expansion assumed in the design. The other values found are also within the range of good practice and may therefore be retained.

17. Calculations for High-Speed Automatic Cut-Off Engine.—As a further illustration, let it be required to find the diameter of cylinder, length of stroke, and rotative speed of a simple, high-speed, non-condensing engine of 100 I. H. P. at rated load, with swinging eccentric, automatic cut-off, and double-ported flat slide valve of moderate length.

By referring to Art. **6,** it will be reasonable to expect a clearance of about 10 per cent. in this case, and, as before, an initial pressure of 100 pounds, gauge, in the cylinder may be assumed. From Art. **11,** it may be assumed that the cut-off will not occur earlier than one-tenth stroke. As the boiler pressure is 100 pounds, gauge, the initial pressure p_1 will be 114.7, and, according to Art. **8,** the back pressure may be taken as 17 pounds, absolute.

Applying formula **1,** Art. **11,** with cut-off at one-tenth stroke, the exhaust must close not earlier than

$$l_s = 1 - \frac{.1\,(114.7 - 17)}{17} = .425 \text{ of the stroke}$$

With a ratio of expansion of 3, from Art. **4,** and a clearance of 10 per cent., the cut-off will come at $\frac{1.1}{3} - .1 = .27$ of the stroke, nearly.

Then, by the provisional valve diagram, as shown in Fig. 3, the exhaust closure is found to be at .562 stroke when the cut-off is at .27 stroke. The use of the Bilgram valve diagram, as explained in *Valve Gears*, Part 1, is recommended for this purpose.

The line $A C$, Fig. 3, is first drawn to represent the stroke of the engine, and the semicircle $A B C$, with O as its center, is drawn to represent the crank-circle. Then, .1 l is laid off to represent the earliest cut-off position of the piston. The perpendicular a and the radial line $O D$, representing the crank position at earliest cut-off, are then drawn. The angle $D O C$ is bisected by the line $O E$, and any point, as O_1, on this bisector is taken as a center for a circle b tangent to $D O$ and $O C$. Next, the earliest point of exhaust closure .425 l is laid off, the perpendicular c erected, and the radial line

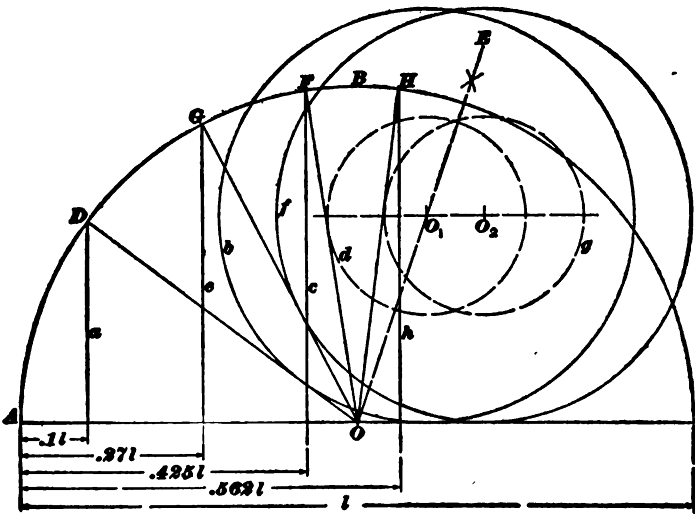

FIG. 3

$O F$ drawn to represent the crank position at exhaust closure. Then, with O_1 as a center, draw the circle d tangent to $O F$. Lay off .27 l on $A C$ to locate the point of cut-off for a ratio of expansion of three, erect the perpendicular e, and draw the crank-line $O G$. Then, on the horizontal line through O_1, locate O_2, so that a circle of the same radius as b will be tangent to $O G$ and $O C$, as circle f; draw circle g with O_2 as a center and of same radius as d, and draw $O H$ tangent to g. Then, drop the perpendicular h, and its distance from A will give .562 l, the point of exhaust closure. No values, except the point of exhaust closure at rated load, should be taken from the provisional valve diagram.

18. Now, from the theoretical indicator diagram, as in the preceding case, the M. E. P. is found to be 52.47 pounds, and the actual M. E. P. will be about $52.47 \times .94 = 49.3$ pounds. Then, by using formula **2**, Art. **16**, and remembering that LN = piston speed, the area of the cylinder is

$$A = \frac{33{,}000 \times \text{I. H. P.}}{PLN} = \frac{33{,}000 \times 100}{49.3 \times 600}$$
$$= 111.6 \text{ square inches, nearly}$$

Correcting for the area of the piston rod, exactly as in the previous case, the area of the free side of the piston is found to be 114 square inches, from which the diameter of the piston and of the cylinder is found to be 12.05 inches. As in the previous case, the diameter would usually be taken as 12 inches.

As the engine is to run at high speed, about 250 may be taken as the number of revolutions per minute. Using this rotative speed and a piston speed of 600 feet per minute, the length of stroke becomes $\frac{600 \times 12}{250 \times 2} = 14.4$ inches. The stroke would then usually be made an even 14 inches, making the rotative speed $\frac{600 \times 12}{14 \times 2} = 257$ revolutions per minute.

19. Calculations for Hoisting and Locomotive Engines.—In the case of hoisting engines, locomotives, and other engines that must start a heavy load from rest, the required diameter of cylinder is determined by the *torque*, or *twisting moment*, on the shaft necessary to start the load. If the cylinder is made large enough to start the load, it will always run the ordinary load at an early cut-off and reduced power. In engines of this class, two cylinders are used.

In order to illustrate, take the case of a hoisting engine, the maximum torque on the shaft of which, in starting the load, is 149,000 foot-pounds. To secure ease in handling the load, stopping, starting, etc., such an engine should be of long stroke—two or three times the diameter of the cylinder—and of slow, rotative speed. Hence, the revolutions per minute may be taken as 65.

20. To insure perfect handling, the engine must have a minimum torque somewhat in excess of 149,000 foot-pounds. Hence, allow 20 per cent. for overcoming friction and accelerating the moving parts. Engines of this type have two cylinders and the cranks are connected at right angles. Thus, it may be assumed with sufficient accuracy, that the minimum torque of the engine will occur when one crank is on the dead center. By neglecting the angularity of the connecting-rod and assuming that the full boiler pressure acts on the piston, the minimum torque of the engine is then the product of the area of one piston, the steam pressure, and the length of the crank. The back pressure may be taken as that of the atmosphere, so that, by taking a boiler pressure of 100 pounds, gauge, the forward pressure in the cylinder will be 100 pounds per square inch of piston area.

Let D be the diameter of the cylinder in inches. Taking the length of the stroke as twice the diameter of the cylinder, the length of the crank in feet is $D \div 12$. Including friction, the torque to be overcome by the engine is $100 \times .7854\, D^2 \times D \div 12$. Equating these values with the starting load of 149,000 with 20 per cent. added gives $78.54\, D^3 \div 12 = 149{,}000 \times 1.2$, or

$$D = \sqrt[3]{149{,}000 \times 1.2 \times 12 \div 78.54} = 30.12, \text{ say } 30, \text{ inches}$$

Hence, the stroke is $30 \times 2 = 60$ inches.

21. Compound and Triple-Expansion Engines. The diameter of the low-pressure cylinder of a compound or triple-expansion engine may be found as in the foregoing cases by assuming that all the work is done in the low-pressure cylinder. In this case, the factor by which to multiply the theoretical M. E. P. to obtain the probable M. E. P. is from .7 to .8 for a compound and .6 to .7 for a triple-expansion engine. As the ratio of the volume of the high-, intermediate-, and low-pressure cylinders is determined, the diameter of the high- and intermediate-pressure cylinders may be found from that of the low-pressure cylinder.

ENGINE DETAILS

CYLINDERS AND STEAM CHESTS

22. Cylinder Proportions.—The proportions here given are largely empirical, being based on an examination of a large number of engines of the classes treated. Fig. 4 illustrates an example of a cylinder designed for a simple slide-valve engine. The crank-end head A is cast solid with the cylinder, while the method of fastening it to the frame B is clearly shown. The thickness of the cylinder walls may be found from either of the following formulas, according to the kind of work the engine is to do:

$$i = .0002pD + .4 \text{ inch} \qquad (1)$$

or,

$$i = .0003pD + .375 \text{ inch} \qquad (2)$$

in which i = thickness of walls of cylinder, in inches;
p = maximum steam pressure, in pounds per square inch, gauge.
D = diameter of cylinder bore, in inches; if p is less than 100 in the actual case, use 100 in the design.

Formula **1** applies only to engines in which lightness is a prime consideration, and in which first-class material and workmanship are assured by careful inspection and tests throughout the whole process of manufacture. Formula **2** applies to cylinders above 10 inches in diameter and gives a value of i much greater than is given by formula **1**. Formula **2** is of general application to ordinary cylinders of slide-valve engines. Cylinders in which the length is great in proportion to the diameter will be treated later.

23. In Fig. 4, the principal dimensions of the cylinder are indicated by letters whose value may be determined by the following formulas, in which D is the diameter of the cylinder in inches:

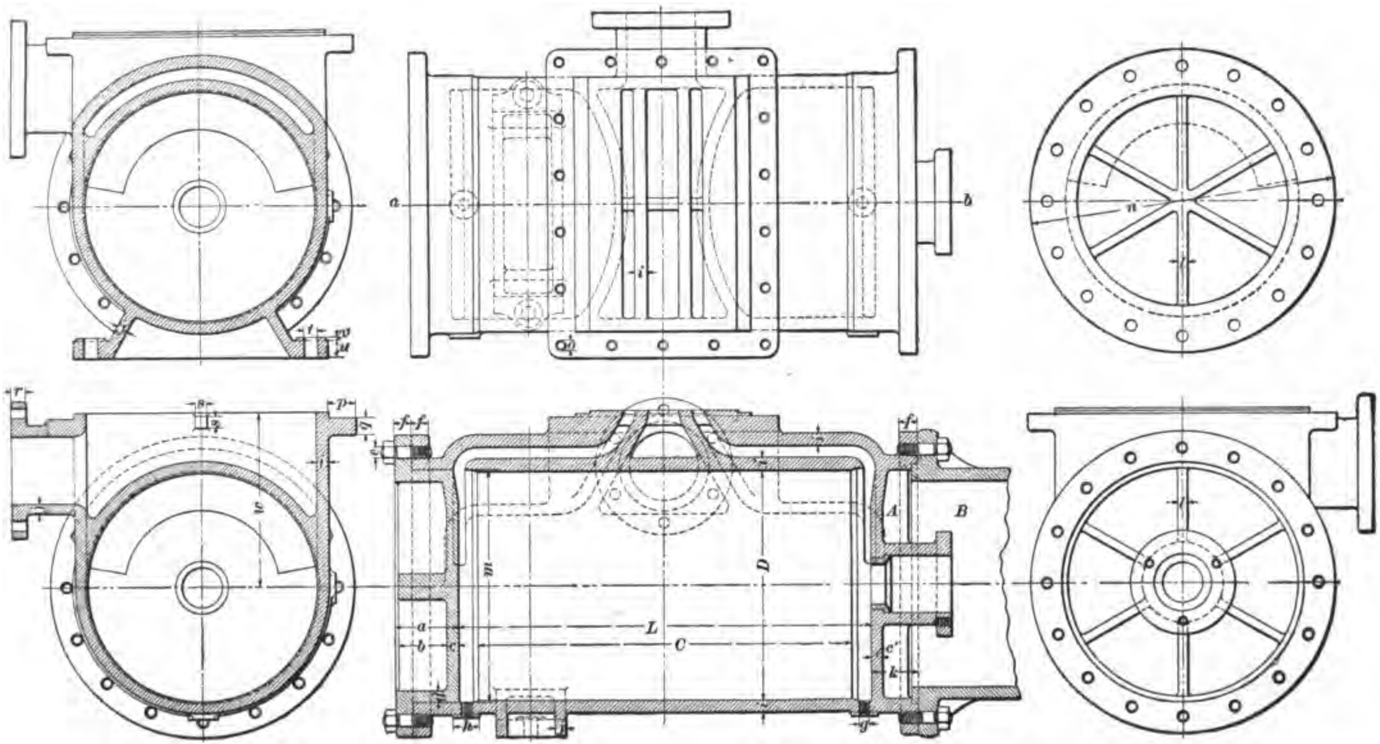

FIG. 4

L = length of stroke + thickness of piston + twice the piston clearance.

C = length between counterbores = length of stroke + distance from outer edge to outer edge of piston rings − $\frac{1}{3}$ width of piston ring.

$a = 5.5\,i$.

$b = 4.2\,i$.

$c = i + .25$ inch.

$c' = 1.1\ (i + .25$ inch$)$.

$d = i$.

e = nominal diameter of cylinder-head bolt or stud. (The method of finding this value is given in the following article.)

$f = 1.5\,i$.

$g = .04\,D + .125$ inch. (Take the nearest nominal size of pipe tap.)

h = twice the outside diameter of drain pipe.

i is found by formula **1** or **2**, Art. **22**.

$j = .85\,i$.

$k = 4\,i$.

$l = .75\,i$.

$m = 1.01\,D + .125$ inch.

$n = m + 6\,e$.

o = nominal diameter of steam-chest cover bolts. (The method of finding this value is given in the following articles.)

$p = 2.75\,o$.

$q = 1.5\,r$.

$r = 1.25\,i$.

$s = i$. (This value is required only when the length of the ports is greater than 12 inches.)

$t = 1.25\,i$; when D is greater than 24 inches, use four bolts in the standard, and make $t = 1.1\,i$.

$u = 1.5\,i$.

$v = .25$ inch.

24. Cylinder-Head and Valve-Chest Cover Bolts. The strength of the cylinder-head and the valve-chest cover bolts or stud bolts holding a cylinder or valve-chest cover, over and above allowances due to screwing up, must be sufficient to hold the cover against ordinary boiler pressure; and, at the same time, the number of bolts must be great enough and close enough together to prevent springing of the flange between bolts, and consequent leakage. A large number of small bolts would tend to secure the latter result; but it must be remembered that the intensity of stress due to screwing up is much more severe in a small bolt than in a large one. Consequently, the bolts used are seldom less than $\frac{3}{4}$ inch in diameter, nor are they often larger than $1\frac{1}{4}$ or $1\frac{3}{8}$ inches in diameter in an engine of ordinary size. The usual range is from $\frac{7}{8}$ to $1\frac{1}{4}$ inches in diameter. Ordinarily, a sufficient number of bolts will be used, so that they will not be farther apart than about 5 inches from center to center around the bolt circle nor more than five times the thickness of the flange in which they are placed, as a steam-tight joint will not be obtained with spacing that is much wider.

The number of bolts is generally determined by convenience in spacing, an even number being used.

Let d = outside nominal diameter of bolt, in inches;
D_1 = diameter, in inches, of cylinder in counterbore;
l = load, in pounds, on each bolt, due to steam pressure;
L = load, in pounds, on each bolt, due to screwing up;
n = number of cylinder-head bolts;
p = maximum steam pressure, in pounds per square inch, gauge;
S_1 = safe tensile stress of material, in pounds per square inch.

The load on each bolt due to screwing up may be taken as

$$L = 16{,}000\,d \qquad (1)$$

The load on each bolt due to the steam pressure is expressed by the formula

$$l = \frac{.7854\,D_1^2 p}{n} \qquad (2)$$

It will be on the safe side to take the total load on one bolt as the sum of the loads due to screwing up and the steam pressure, or $L + l = 16{,}000\,d + \frac{.7854 D_1^2 p}{n}$. The safe tensile stress in one bolt is $.7854\,d^2 S_1$, approximately; hence, by placing this value equal to the total load on one bolt, and solving for d, the following formula is obtained:

$$d = \frac{10{,}186}{S_1} + \sqrt{\frac{D_1^2 p}{S_1 n} + \left(\frac{10{,}186}{S_1}\right)^2} \qquad (3)$$

In good practice a factor of safety of about 3 is commonly used. With small bolts, it may be necessary to allow a factor as low as 2.9, while with large bolts, a factor of about 3.3 may be obtained. This makes the safe tensile stress in the bolts from about 15,000 to 17,000 pounds per square inch for wrought iron, and from 18,000 to 21,000 for steel.

The nominal outside diameter of a bolt or stud of either wrought iron or steel should not be less than $\frac{3}{4}$ inch. Having found the required diameter by the foregoing formulas, the nearest standard size should be used.

25. The number and diameter of the bolts in the steam-chest, or valve-chest, cover may be found in a manner similar to that employed for the cylinder head. The bolts should not be less than $\frac{3}{4}$ inch in diameter, nor spaced farther apart than 5 inches between centers.

Let a_1 = length of valve-chest cover, in inches, between the center lines of bolts on opposite ends;

b_1 = breadth of valve-chest cover, in inches, between the center lines of bolts on opposite sides.

Then, with the other quantities the same as in Art. **24**, the load on each bolt due to screwing up is

$$L = 16{,}000\,d \qquad (1)$$

The load on each bolt due to the steam pressure is

$$l = \frac{a_1 b_1 p}{n} \qquad (2)$$

Placing the sum of the loads in formulas **1** and **2** equal to the safe stress on one bolt gives $.7854\,d^2 S_1 = 16{,}000\,d + \frac{a_1 b_1 p}{n}$, which reduces to

$$d = \frac{10{,}186}{S_1} + \sqrt{\frac{a_1 b_1 p}{.7854 S_1 n} + \left(\frac{10{,}186}{S_1}\right)^2} \qquad (3)$$

The safe tensile stress would be from about 15,000 to 17,000 pounds per square inch for wrought iron, and from 18,000 to 21,000 for steel.

26. Steam Ports and Passages.—The dimensions of steam ports, exhaust ports, and other steam passages depend on the velocity of flow of the steam. The ports and passages must be large enough to permit the steam to follow up the advancing piston without loss of pressure.

The area of cross-section of the steam and exhaust pipes may be found from the formula

$$a = \frac{A v}{V}$$

in which a = area of cross-section of the pipe, in square inches;
A = area of piston, in square inches;
v = piston speed, in feet per minute;
V = velocity of flow of steam in the pipe, in feet per minute.

The values of V commonly used in practical construction are 4,000 for the exhaust pipe, and 6,000 for the steam pipe, although these values are sometimes increased to as much as 6,500 and 8,000, respectively. As the pipes are circular, the internal diameters can readily be obtained from their areas of cross-section. The nearest standard size of pipe to that found in the manner just described should be used.

The area of cross-section of the steam and exhaust ports may also be found from the formula just given. The values of V used in practical construction are from 4,000 to 6,000 for the exhaust port and from 6,000 to 8,000 for the steam port, the lower values being the most common. Where the same port is used for both steam and exhaust, the area of the port must be made large enough for the exhaust. If l is the length of the port and D the diameter of the cylinder, then l is made equal to about $.7\,D$ to D, the usual value being between $.8\,D$ and $.9\,D$ for slide-valve engines, and about $.9\,D$ to D for the Corliss type.

With steam superheated about 100° F., the values of V just given may be increased from 30 to 40 per cent. in the case of the steam pipe, and if separate ports are used for steam and exhaust, the value of V for the steam port may also be increased the same amount.

The height w, Fig. 4, of the valve seat above the center line of the cylinder should be made as small as possible without interfering with the size of the steam and exhaust ports.

27. Steam-Chest Calculations.—In Fig. 5 is shown a design of a cylinder having the steam chest cast solid with it. The crank-end head in this case is a separate cast-

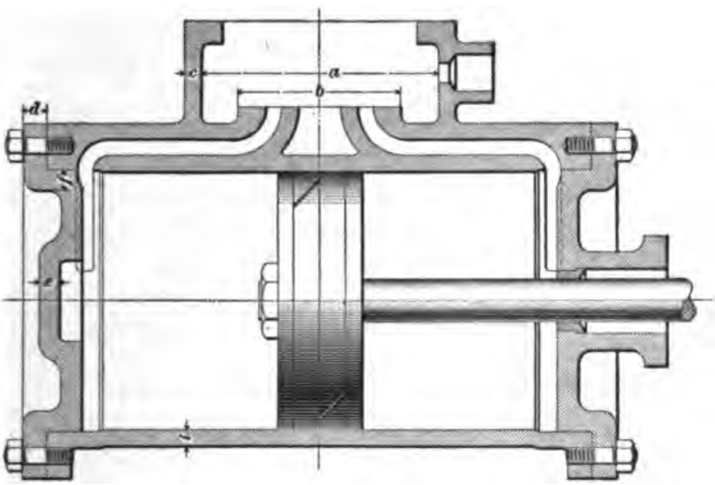

FIG. 5

ing fitted to the cylinder in the same manner as the head-end head. The heads, which are cast without ribs, are well suited for cylinders of small diameters. For larger diameters, the ribbed heads shown in Fig. 4 are better.

The following proportions apply to Fig. 5:

p = maximum steam pressure, in pounds per square inch, gauge.

D = diameter of cylinder, in inches.

$i = .0003\,p\,D + .375$ inch.

a = length of valve + travel of valve + twice the clearance between valve and steam chest at ends of valve travel.
b = valve travel + length of valve $- \frac{1}{4}$ to $\frac{1}{2}$ inch.
$c = i$.
$d = 1.5\,i$.
$e = 1.25\,i$.
$f = 1.25\,i$.

All other dimensions are to be determined by the formulas given for Fig. 4.

28. Fig. 6 illustrates a steam chest for the cylinder

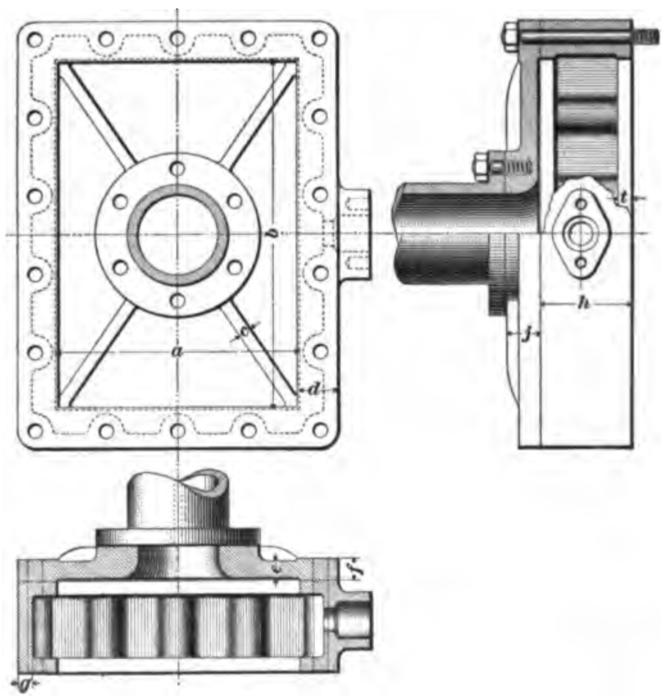

FIG. 6

shown in Fig. 4. The principal dimensions are to be determined by the following proportions, which are based on the thickness i of the cylinder walls, as found in Art. **22**, and on the travel and dimensions of the valve:

a = length of valve + travel of valve + twice the clearance between the valve and the steam chest at ends of valve travel.

b = breadth of valve + twice the clearance between one end of the valve and the steam chest.

c = $.75\,i$.

d = $2.75\,o$, where o is the nominal diameter of the steam-chest bolts.

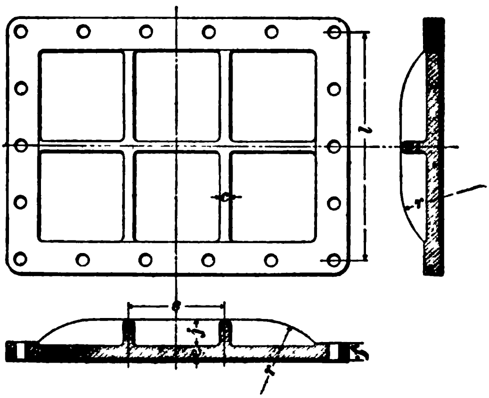

FIG. 7

e = $.05\sqrt{A'} + .125$ inch, in which A' is the area of steam-chest in square inches, obtained by multiplying the length between center lines of bolt rows by the width between center lines; but e should never be made less than i, whatever may be the result of calculation by this formula. When the steam pressure is above 100 pounds per square inch, gauge, the dimension should be changed to $e = .05\sqrt{\frac{p\,A'}{100}}$ + .125 inch.

f = $1.3\,e$.

g = i.

h = height of valve + necessary clearance.

j = $2.5\,i$. When the area of the steam-chest cover exceeds 600 square inches, the height of the ribs should be $3.5\,i$, and their number should be increased.

t = $.85\,i$.

29. Fig. 7 shows a steam-chest cover design that should be used when the steam-pipe flange is to be located on one side of the steam chest. The dimension indicated by the letters have the following values:

e = $.05\sqrt{A'} + .125$ inch for the thickness of the cover, as in Art. **28.**

$c = e$.

$f = 1.3\,e$.

$j = 2.5\,e$, at least.

l = width of steam-chest cover between center lines of bolts, in inches.

p = steam pressure, in pounds per square inch, gauge.

$r = 7.7\,e$.

s = distance between centers of ribs, and should never exceed the distance in inches given by the formula

$$s = \sqrt{\frac{40\,e_1^2}{p}}, \qquad (1)$$

in which e_1 is the thickness of the cover in sixteenths of an inch.

To insure safety, the shorter ribs across the cover should be sufficiently strong to carry the total load and at the same time allow a small factor of safety, say about 2, without any aid from the long rib and the flat part of the cover. To obtain this result, let

$$S_1 = \tfrac{3}{4}\frac{p\,s\,l^2}{c\,(e+j)^2}, \qquad (2)$$

in which S_1 is the safe stress of cast iron in flexure, in pounds per square inch.

Substituting in formula **2** the values of the letters as just stated, S_1 should not exceed 15,000; if it does, either the thickness or the height of the ribs, or both, should be increased, or their distance apart should be decreased until the resulting value of S_1 does not exceed 15,000.

EXAMPLE.—Find the thickness of the cover and the thickness, height, and pitch of the ribs of the cover for a steam chest having a maximum length and width of 24 and 16 inches, respectively. The chest is subjected to a steam pressure of 160 pounds per square inch.

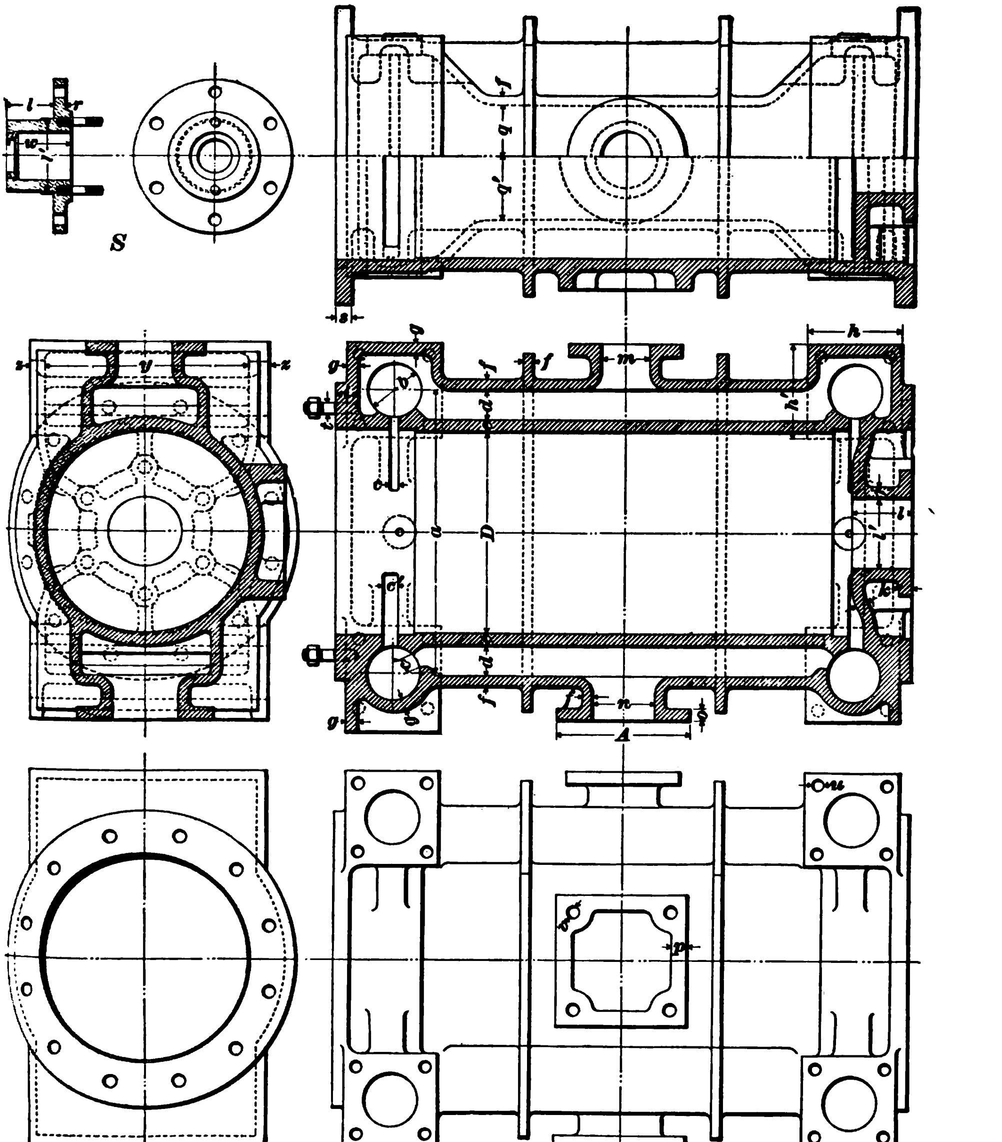

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FIG. 8

SOLUTION.—The thickness of the cover is $e = .05\sqrt{24 \times 16} + .125$ in. $= 1.105$ in. Use $e = 1.125$, or $1\frac{1}{8}$, in.

The thickness of the ribs on the cover is $c = 1\frac{1}{8}$ in.

The height of the ribs above the cover is $j = 2.5 \times 1.125 = 2.81$ in. Use $j = 2\frac{7}{8}$ in.

Then, applying formula **1** for the pitch of the ribs,

$$s = \sqrt{\frac{40 \times (18)^2}{160}} = 9 \text{ in.}$$

Use two short ribs 8 in. apart and one long rib over the middle of the chest. Then, in order to be certain that the cover will be safe, apply formula **2** to find the stress in the ribs. Thus,

$$S_1 = \frac{3}{4} \times \frac{160 \times 8 \times (16)^2 \times 8}{9 \times 4 \times 4} = 15{,}360 \text{ lb.},$$

which is satisfactory.

30. Corliss-Engine Cylinder Proportions.—Corliss and other long-stroke engines, especially those having steam passages along one side and exhaust passages along the other side of the cylinder, should be designed according to the expressions given here rather than by those in Arts. **22** and **23**. Fig. 8 shows a Corliss-engine cylinder, which may be designed according to the following proportions:

D = diameter of cylinder, in inches.

$a = 1.21\,D + 2e + .125$ inch for double-ported engines, and $1.26\,D + 2e + .125$ inch for single-ported.

$b = .2\,D$ for double-ported engines, and $.25\,D$ for single-ported.

c, the width of the steam port, should be determined according to Art. **26**.

c', the width of the exhaust port, should also be determined according to Art. **26**.

$d = .17\,D$.

$e = .0005\,p\,D + .375$ inch for cylinders up to 30 inches in diameter; beyond 30 inches, e may be reduced to $e = .0004\,p\,D + .375$ inch. In the formula for e, p is the boiler pressure in pounds per square inch, gauge. The value of P in this formula, however, should never be taken less than 100.

$f = e$.

$g = e$.

$h = b + 2\,(c + g)$.

$h' = h$.

$i = 1.8e$.

$j = e$.

$k = 1.2e$.

$l = 1.7x - 1.2e + 2$ inches, where x is the diameter of piston rod, in inches.

$l' = .32D$, about.

m should equal the diameter of the steam pipe, as designed according to Art. **26.**

n should equal the diameter of the exhaust pipe, as designed according to Art. **26.**

$o = 1.25e$.

$p = 1.3e$.

$q = .25D$.

$q' = .323D$.

$r = 1.2e$.

$s = 1.5e$.

t = diameter of bolts, and should be found according to Art. **24** for cylinder-head bolts.

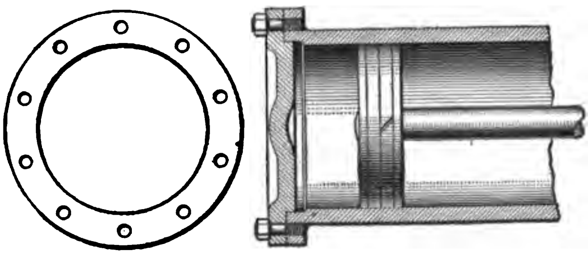


FIG. 9

u = diameter of bolts for the steam-valve chests, and may be found according to Arts. **24** and **25** for valve-chest cover bolts. It is best to use bolts of the same size on the exhaust-valve chests.

$v = 1.2e$; take nearest standard-sized bolt.

$w = 1.7x + 2.25$ inches, where x is the diameter of piston rod.

$y = .9D$ to D.

$z = 1.5e$.

Where pipes are connected, the diameter and spacing of the bolts should be made to suit the piping to be attached. In this cylinder, the stuffingbox S is a separate piece that is to be bolted to the cylinder head.

Fig. 9 shows a cylinder head suitable for cylinders of small diameter; its thickness may be made equal to the thickness of the cylinder + .25 inch.

ENGINE SHAFTS AND CRANKS

THE SHAFT

31. Diameter of Shaft.—The general dimensions of crank-shafts are computed according to the principles of machine design. The calculations are made with regard to all the principal stresses that are likely to come on the shafts. The simplest formula is that in which only the torsion of the shaft is considered, when the formula is reduced to

$$d = 68.45\sqrt[3]{\frac{H}{N S_s}}, \qquad (1)$$

in which d = diameter of shaft, in inches;
H = I. H. P. at rated load;
N = number of revolutions per minute;
S_s = safe shearing stress of material in shaft.

When there are forces tending to bend the shaft, as the weight of a flywheel, they must also be taken into consideration, and the formula then becomes more complicated.

When a series of different-sized engines of the same type are to be built, it may be assumed that they will run under about the same conditions. In such a case, it is unnecessary to use the general method of calculating the crank-shaft just given, for short empirical formulas may be deduced from the practice of the best makers.

An examination of a large number of stationary side-crank engines, all low-speed, shows the following relation between d, H and N:

$$d = K\sqrt[3]{\frac{H}{N}}, \qquad (2)$$

in which K varies from 5.66 to 7.8, the average in good practice being about 6.36.

The maximum stresses occur in the shaft in which K has the smallest value, and the minimum where K has the largest value. These stresses vary from 17,750 to 6,780 pounds per square inch, and take into consideration both torsion and bending.

32. Journal of Engine Shaft.—Certain dimensions of journals are necessary for the cool running of engine shafts.

Let l = length of main journal, in inches;
H = I. H. P. at rated load;
L = length of stroke, in inches;
d = diameter of shaft at bearing, in inches;
D = diameter of cylinder, in inches.

An examination of a large number of stationary side-crank engines, all low-speed, shows the following relations:

$$l = K_1 \frac{H}{L} + 7, \qquad (1)$$

in which K_1 varies from .86 to 2.27, the average being about 1.56.

$$d = K_2 \frac{D^2}{l}, \qquad (2)$$

in which K_2 varies from .36 to .5, the average being about .44.

The product of d and l is called the *projected area* of the journal or bearing, and the pressures per square inch of projected area found vary from 156 to 218 pounds, 178 pounds being the average. Some authorities recommend values as high as 450 pounds per square inch, but an investigation of a large number of engines in use shows the values just given. The diameter of the shaft must be calculated for both strength and cool running, and the larger values of d found should be used.

33. An examination of stationary center-crank engines, all high-speed, shows the following relations:

$$d = K_3 \sqrt{\frac{H}{N}}, \qquad (1)$$

For cool running, the length of a journal, such as a crank-pin, should theoretically be given by the following formula, in which K is a constant:

$$l = \frac{KH}{L} \qquad (1)$$

An examination of the proportions used in practice shows that the length is given more closely by an expression of the form

$$l = K\frac{H}{L} + K_1 \qquad (2)$$

An examination of side-crank, low-speed stationary engines shows that the value of K_1 is equal to 2 in all cases, and that in good practice K varies from .345 to .655.

For cool running, the following formula applies in all cases:

$$d' = .7854\frac{D^2 p}{p_1 l} \qquad (3)$$

Some authorities place the value of p_1 between 800 and 900 for low-speed, side-crank stationary engines, but an examination of actual engines shows that this value in good practice runs from 873 to 1,570.

35. For strength, the following formula for the diameter may be used in all cases:

$$d' = \sqrt[3]{\frac{4pD^2l}{S_1}}, \qquad (1)$$

in which S_1 is the safe tensile stress. With the low-speed, side-crank stationary engine, S_1, in good practice, varies from 3,200 to 12,500, the average value being about 7,000.

When p and S_1 are assigned numerical values, formula **1** may be reduced to the expression

$$d' = K_2\sqrt[3]{D^2 l}, \qquad (2)$$

in which K_2 represents the factor $\sqrt[3]{\frac{4p}{S_1}}$.

An examination of low-speed, side-crank stationary engines shows that in good practice K_2 varies from .32 to .5, with .384 as the average value.

36. For stiffness, the following formula gives the value of d':

$$d' = .016\sqrt[4]{\frac{p D^2 l^3}{k}} \qquad (1)$$

In low-speed, side-crank stationary engines, the deflection k, in good practice, varies from .0004 to .005, the average value being about .001.

Where p and k have numerical values, the formula just given reduces to the form

$$d' = K\sqrt[4]{D^2 l^3} + K_1 \qquad (2)$$

An examination of low-speed, side-crank stationary engines shows that, in good practice, K_1 varies from .6 to 1, with an average value of .81, and that K varies from .185 to .353, with an average value of .28.

For cool running, for strength, and for stiffness, the diameter of the pin should be found independently, and the largest value should be used. Generally, this value will be given by the calculations for stiffness, but this should not be taken for granted, and all three calculations should be made.

37. Crankpin for Center Crank.—Center crankpins are found to have very different dimensions than overhung pins. For cool running, instead of using the theoretical expression, the length of the pin is found to be given more closely in practice by the formula

$$l = K\frac{H}{L} + K_1 \qquad (1)$$

An examination of center-crank, high-speed stationary engines shows that when $K = .35$ and $K_1 = 2.2$ inches, formula **1** gives the average value of l in good practice; but many pins show wide departures from average practice.

For cool running, the diameter for all cases is found by the formula

$$d' = .7854\frac{D^2 p}{p_1 l} \qquad (2)$$

With high-speed, center-crank stationary engines, p_1 varies from 225 to 1,400, the average being about 450. With marine engines, p_1 varies from about 400 to about 600.

Where p and p_1 have definite values, formula **2** reduces to

$$d = K\frac{D^2}{l}, \qquad (3)$$

in which K represents $\frac{.7854p}{p_1}$.

An examination of high-speed, center-crank stationary engines, shows that K varies from .089 to .599, with an average value of .28 in good practice.

A purely theoretical consideration of the strength and stiffness of the center-crank pin would indicate that it might be of smaller diameter than the shaft, but in practice this is found to be unsatisfactory. The diameters of center-crank pins in actual engines are always about the same as those of the main journals, being sometimes a little less, but greater as a rule. It may therefore be considered an empirical rule, sufficiently established by practice, that a center-crank pin should be of about the same diameter as the main journals—better greater than smaller. If the diameter necessary for cool running is greater than the diameter of the main journals, this greater diameter should be taken.

CRANK AND COUNTERBALANCE

38. Fig. 11 shows a style of crank much used on low-speed engines, such as the Corliss type. Let S be the maximum stress on the material, in pounds per square inch; and r the length of the crank, in inches. The other symbols represent the same quantities as given in Arts. **31** to **37**. The dimensions are to be computed in the following manner:

For d, use the method explained in Art. **31**.

a = .75 to 1 times the diameter of the shaft. An effort should be made to keep down the value of a, as by so doing the bending moment on the main bearing is decreased.

To obtain sufficient strength opposite the key shown in Fig. 11, b should be found by the following formula:

$$b = .75d + .5\sqrt{\frac{d^2}{4} + \frac{1.57D^2pr}{aS}} \qquad (1)$$

S may be taken as 9,000 for wrought iron and 11,000 for steel; p is the maximum unbalanced steam pressure, but should never be taken less than 113 pounds per square inch.

After finding b by means of the formula just given, the stress in the boss at right angles to the crank should be found by the formula

$$S_4 = \frac{4.71\,D^2 p r b}{a(b^3 - d^3)} \qquad (2)$$

If S_4 as thus calculated exceeds the safe bending stress for the material, b should be increased until S_4 is sufficiently reduced.

$c = .045\,d + .0625$ inch.

d' should be calculated as explained in Arts. **34** to **36.**

$f = .375g$, as a trial value. If desirable, this may be increased later.

g is found by drawing lines tangent to the circles b and i, and taking the distance between these lines at a point midway between the main shaft and crankpin centers. The stress at g should then be found by the formula

$$S_4 = \frac{2.36\,D^2 p r}{g^2 f} \qquad (3)$$

As before, p should not be taken less than 113. If the value of S_4 as thus found is beyond the safe bending stress of the material, f should be increased. If f cannot be increased enough to bring the stress down to a safe value, b and i should be increased.

$h = 1.35\,d'$.

As a trial value, i may be made $2\,d'$; after f and g are found, i may be calculated by the formula

$$i = \frac{.7854\,D^2 p}{f S_1} + d' \qquad (4)$$

The remarks in regard to the use of p and S_4 given in connection with formula **3** apply equally as well to p and S_1 in this formula. S_1 is the safe tensile stress in the material.

l should be calculated the same as in Art. **34.**

39. Modern high-speed engines are often counterweighted to reduce vibrations in running, and the crank in such cases usually takes the form of a disk, as shown in Fig. 12. The disk is hollowed out as shown, but a portion of the material is left in the side opposite the crankpin so as to form the counterweight, which, by its centrifugal force, counteracts the centrifugal force resulting from the motion of the reciprocating and other rotating parts of the engine. As here shown, the counterweight may be made as a separate part from the disk proper, it being attached only to the hub *a*, with which it is cast in one piece; it is thus made to allow for expansion and contraction in the larger crank-disks. The

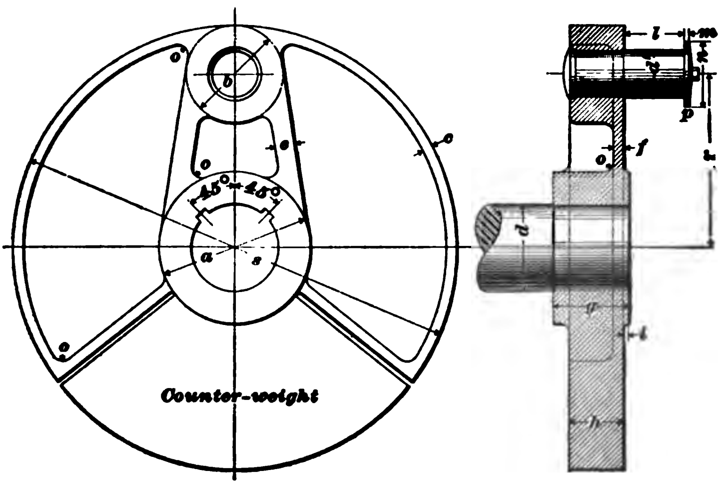


FIG. 12

width of the split is about $\frac{3}{4}$ inch for engines of 48-inch stroke or less, and 1 inch for all larger sizes.

In Fig. 12, *p* is a plate held to the end of the crankpin by a tap bolt. The radius of all fillets except that of the boss *i* is *o*. Assume that *s* is the diameter of the crank-disk in inches; *r*, the length of the crank-arm, in inches; *D*, the diameter of the cylinder, in inches; *p*, the maximum unbalanced pressure on the piston, in pounds per square inch, which should never be taken as less than 113; *S*, the maximum stress on the material, in pounds per square inch; and *k*, the deflection of the outer end of the pin, in inches. The significance of all other symbols used in this article is shown in Fig. 12. The different dimensions should be determined from the following rules:

a should be determined by formula **1**, Art. **38**, for finding *b* of Fig. 11, but the calculation of the stress in the section of the hub at right angles to the center line of the crank, as found in formula **2**, Art. **38**, for Fig. 11, may be omitted.

b should be calculated by formula **4**, Art. **38**, for finding *i* of Fig. 11.

$$c = \frac{r}{24} + .5 \text{ inch}$$

d' should be calculated for cool running by the formulas of Art. **34**. If the rotative speed is above 125 revolutions per minute, the formulas of Art. **37** for high-speed center-crank engines should be used. If the rotative speed is not greater than 125 revolutions per minute, the formulas of Art. **34** for low-speed side-crank engines should be used.

For strength, use formula **1**, Art. **35**. S_1 may be taken as 9,000 for wrought iron, and 11,000 for steel.

For stiffness, use formula **1**, Art. **36**, taking the deflection *k* as .001, as in Art. **34**.

d' should be calculated separately for cool running, strength, and stiffness, and the greatest value found should be taken.

$e = 2c$.

$f = c$.

g should be somewhat greater than *h*, and is usually about .875 *d*.

h = thickness of counterweight, to be calculated so as to make the counterweight as heavy as necessary.

i should be given such a value that the connecting-rod will have about ⅛-inch clearance.

l should be calculated by the formulas given in Arts. **34** or **37**. If the rotative speed is above 125 revolutions per minute, the formulas of Art. **37** for high-speed center-crank engines should be used. If the rotative speed is 125 revolutions per minute or less, the formulas of Art. **34**, for low-speed side-crank engines should be used.

$m = .045\,d + .0625$ inch.
$n = 1.35\,d'$.
$o = c$.
$s = 2\,r + b$, for the crank-disk proper and generally for the counterweight also, but, if necessary, the value of s for the counterweight may be increased or diminished in order to get the proper amount of counterweight. This method is objectionable, however, and should be avoided if possible.

40. Counterbalancing the Crank.—In order to find the thickness of the counterweight used on a crank like the one shown in Fig. 12, it is first necessary to find the weight.

Let W_1 = weight of counterweight, in pounds;
W_2 = weight of crank, outside of hub around main shaft, in pounds;
W_3 = weight, in pounds, of the reciprocating parts; that is, weight of piston + weight of piston rod + weight of crosshead + one-half the weight of connecting-rod;
X = distance of center of gravity of counterweight from center of shaft;
Y = angle between the two radial sides of counterweight, this angle being always equally divided by the center line of the crank.

Then, the distance X is found by the formula

$$X = \frac{38.2\,(s^3 - a^3)\sin\frac{Y}{2}}{(s^2 - a^2)\,Y} \qquad (1)$$

When the angle Y is known in degrees, then the formula may be written

$$X = K\frac{s^3 - a^3}{s^2 - a^2} \qquad (2)$$

Thus, when the angle $Y = 60°$, $K = .32$; when $Y = 90°$, $K = .3$; when $Y = 120°$, $K = .276$; when $Y = 180°$, $K = .21$.

To counterbalance so as to avoid knocks when the crank is at right angles to the center line of the engine, the weight of the counterbalance is found by the formula

$$W_1 = \frac{W_3\,r}{X} \qquad (3)$$

To counterbalance so as to avoid knocks when the crank passes over dead center, the counterweight is found by the formula

$$W_1 = \frac{(W_2 + W_3)\,r}{X} \qquad (4)$$

For the greatest possible smoothness of running throughout the revolution, use the formula

$$W_1 = K\frac{(W_2 + W_3)\,r}{X}, \qquad (5)$$

in which K has a value of from .67 to .75, the smaller value being used where it is desired to have absence of vibration when the crank is on the quarter, that is, at right angles to the center line of the engine, and the larger value where absence of vibration is most desirable when the crank is at dead center.

In general, the counterbalancing of vertical engines should tend toward the greatest possible smoothness of running, with the crank on the quarter; and that of horizontal engines, with the crank on dead center.

As the counterweight is usually made of cast iron, the thickness h for this material may be found by the following formula, in which Y is in degrees, s and a in inches, W_1 in pounds, and the weight of cast iron is taken as .26 pound per cubic inch:

$$h = 1{,}763\,\frac{W_1(s^2 - a^2)}{Y} \qquad (6)$$

THE PISTON

PISTON BODY

41. Hollow Pistons.—Engine pistons are made in a great variety of forms. For small engines, that is, for cylinder diameters less than 8 or 10 inches, the piston is often a solid disk of cast iron or steel. A form of **hollow piston** that is much used for small engines is shown in Fig. 13, and consists simply of a hollow circular disk of cast iron. The packing rings s, s are made of cast iron and are split and sprung into place; their elasticity causes them to press against the cylinder walls and thus prevents the leakage of steam. The following proportions will give dimensions suitable for this piston:

D = diameter of cylinder, in inches.

For high-speed engines, $a = .3\,D$, as minimum value; $.46\,D$, as average value; and $.6\,D$, as maximum value.

For low-speed engines, $a = .25\,D$, as minimum value; $.32\,D$, as average value; and $.45\,D$, as maximum value.

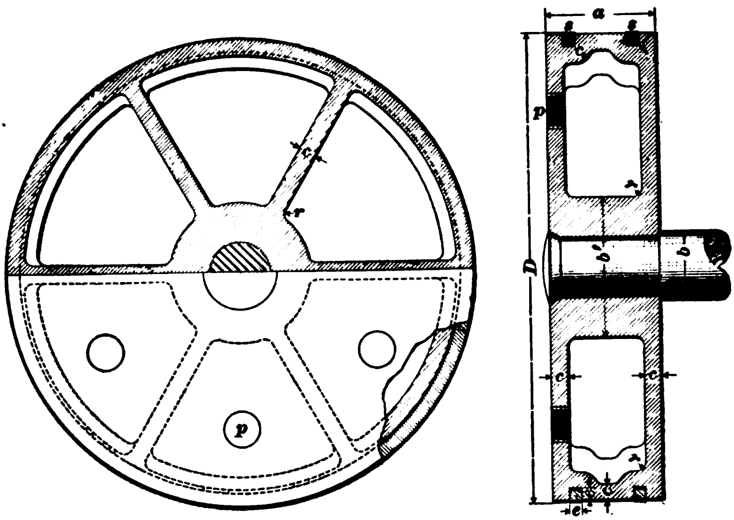


FIG. 13

Both the stiffness and the strength of the piston are greatly increased by increasing the value of a; on the other hand, the weight is increased, and, in horizontal engines, the friction between the piston and the cylinder is also increased.

b = diameter of piston rod.

$b' = 2\,b$.

$c = .18\sqrt{2\,D}$.

$e = .75\,c$.

$r = .5\,c$.

p is a core plug of suitable diameter.

Number of ribs $= .08\,(D + 34)$.

42. Built-Up Pistons.—The piston shown in Fig. 14 is made in two parts; the main part A is called the *spider*, to which is bolted the *follower plate* B. The spider is cast hollow, with radiating arms and lugs for the follower bolts i. A split cast-iron *bull ring* b is placed around the spider, and is supported by steel springs e, which are in turn supported by brass studs f. The bull ring forms a support for the packing rings s, s.

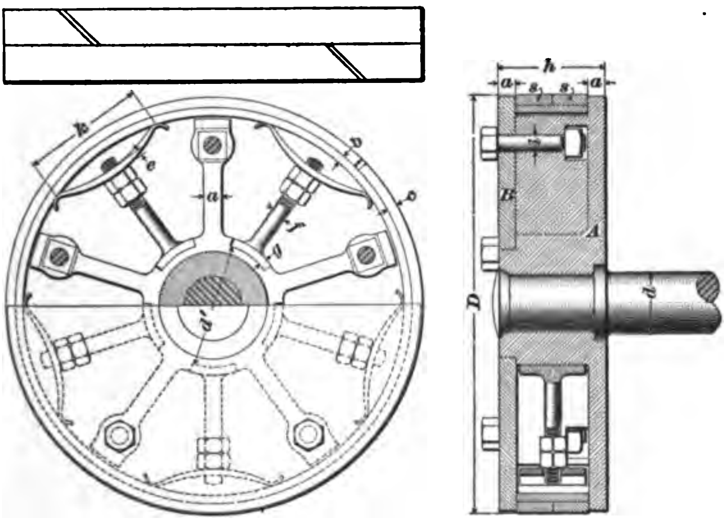


FIG. 14

The dimensions of this piston are given by the following proportions:

D = diameter of cylinder, in inches.

$a = .18\sqrt{2D}$.

$b = .45a$.

$c = .65a$.

d = diameter of piston rod.

$d' = 2d$.

$$e = \frac{.06D}{n}.$$

$f = a$.

$g = .5f$.

h should be calculated by the rules given for a in Art. **41.** In the built-up piston, h should tend rather toward the maximum of practice, as a follower plate does not contribute nearly so much to the strength of the piston as does a solid back cast on the piston. The strength of the built-up piston is derived chiefly from the radiating arms a of the spider, the strength of which is greatly increased by increasing their depth.

$i = .1\left(\frac{D}{50}\sqrt{P} + 1\right) + .25$ inch, where p is the maximum unbalanced pressure on the piston, in pounds per square inch, which should never be taken as less than 113.

$k = 1.4\,\frac{D}{n}.$

n = number of ribs $= .08(D + 34)$.

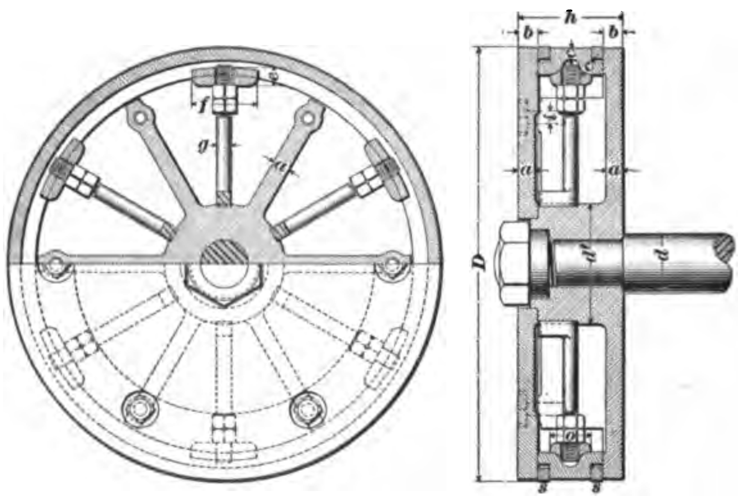


FIG. 15

43. Another form of built-up piston is shown in Fig. 15. The proportions to be used for this piston are:

D = diameter of cylinder, in inches.

$a = .18\sqrt{2D}.$

$b = 1.25\,a.$

$c = .75\,a.$

d = diameter of piston rod.
$d' = 2d$.
$e = 1.25a$.
$f = 3.75a$.
$g = a$.
h should be calculated by the rules given for a in Art. **41.** In the built-up piston, the depth h should tend rather toward the maximum of practice, as the follower plate does not contribute nearly so much to the strength of the piston as does a solid back cast in one piece with the piston.

$i = a - .125$ inch, but should never be reduced below .75 inch.

$o = 2.5a$.
n = number of ribs $= .08(D + 34)$.

44. Solid Pistons.—A form of piston much used in locomotive practice is shown in Fig. 16. A piston of this

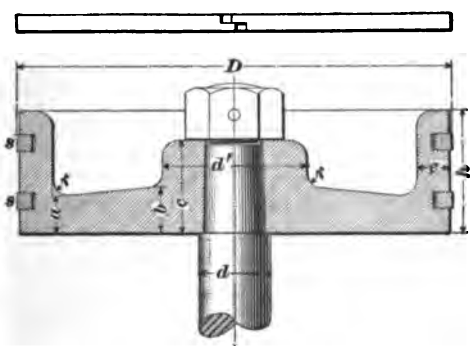


FIG. 16

kind may be made of cast iron, but it is usually a steel casting. Suitable proportions for a cast-iron piston are:

D = diameter of cylinder, in inches.
$c = .15D + 1.125$ inches.
d = diameter of piston rod.
$d' = 2d$.
$e = .27\sqrt{2D}$.
$h = .2D + 1.5$ inches.
$r = .5e$.

The thickness of the body, or disk, of the piston at any point between the boss and the flange should be calculated by the following formula:

$$T = \sqrt{\frac{3p\left(\frac{D^2}{4} - \frac{x^2}{3}\right)}{S}},$$

in which T = thickness, in inches, of piston disk;

p = maximum unbalanced pressure on piston, in pounds per square inch;

D = diameter of cylinder, in inches;

x = distance, in inches, from center of piston to point at which thickness is to be found;

S = maximum stress in material at point, which should not be taken as more than 3,000 pounds per square inch for cast iron.

45. If the piston is to be a steel casting, the following proportions should be used, all other dimensions being the same as for a cast-iron piston:

$d' = 1.75\,d.$

$e = .23\sqrt{2\,D}.$

The thickness of the body, or disk, of the piston should be calculated by the same formula as for cast iron, but S may be given a value as high as 5,500 pounds per square inch.

In either case, it is preferable to make the front of the piston flat, as shown in Fig. 16. The thickness at the point a just at the inner edge of the fillet between the disk and the flange and the thickness b just at the outer edge of the fillet between the boss and the disk should be calculated by the formula just given. These thicknesses should then be laid off on the drawing from the flat face of the piston. A straight line drawn between the two points of the back thus determined will give the outline of the rest of the back of the disk.

Since the piston is solid, the rings must be cut and sprung into place. In order to make the clearance small, the form of the cylinder head is made to conform to the piston.

46. Marine Pistons.—Fig. 17 shows a piston made of a steel casting designed for the high-pressure cylinder of a marine engine. The dotted lines show how the cylinder head is made in order to fit closely around the piston. A conical form is given the piston in order to increase its strength.

The following proportions will give suitable dimensions for this piston for cylinders from 7 to 50 inches in diameter:

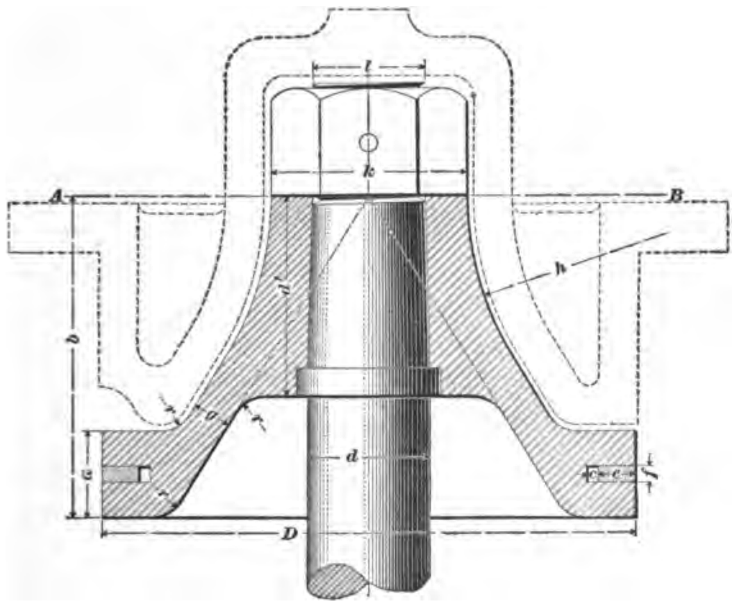


FIG. 17

D = diameter of high-pressure cylinder, in inches.
$a = .1\,D + 1.5$ inches.
$b = \sqrt{20\,D} - 7.5$ inches.
$c = .1\,\sqrt{D}$.
d = diameter of piston rod.
$d' = 1.63\,d$. This expression gives values of d', which are amply large.

The use of a large boss has its own dangers. In casting, the body of metal produced in the center of the piston may remain fluid after the rest of the piston has set, and then, on cooling, may produce permanent stresses in the piston or even tear it apart. The same is true wherever a sudden great variation of thickness is introduced in a casting; therefore, this condition should always be avoided if possible. The designer must depend on his knowledge of foundry practice to avoid such dangers; and whenever he is in doubt it is advisable to consult the foundry foreman.

47. If the boss of the piston in Fig. 17 should appear to be too heavy to cast well, its depth may be reduced; in good practice, d' is often made as small as $.09\,D$. When d' is very much reduced, the shearing stress on the boss should be calculated, and should not exceed 5,000 pounds per square inch.

$e = .225\sqrt{D} + .625$ inch.

$f = .1\sqrt{D} + .25$ inch.

$g = .07\,D$.

h is found by trial, with the center on the line AB.

$k = 1.73\,l + .1875$ inch.

l = diameter of threaded end of piston rod.

$$r = \frac{(a - f)}{2}.$$

48. In Fig. 18 (a) is shown a piston suitable for the intermediate and low-pressure cylinders of the engine for which the piston shown in Fig. 17 is designed.

To compute the dimensions for this piston, make the dimensions $a, b, c, d, d', e, f, k, l$, and r, the same as for the high-pressure piston in Fig. 17. If the engine is compound, let D' represent the diameter of the low-pressure cylinder in inches and make $s = .02\,D'$. If the engine is of the triple-expansion type, let D' represent the diameter of the intermediate cylinder and compute s as before, using the value found for both intermediate and low-pressure pistons. Make the remaining dimensions as follows:

$m = 1.8\,d$.

n is to be found by trial, with the center on the line representing the bottom edge of the piston.

o is to be found by trial, with the center on the line AB.

$t = .2\sqrt{2\,D}$ and $t' = .275\sqrt{2\,D}$, in which D is the diameter of the cylinder for which the piston is made.

The formulas given for the thickness of the body of the piston in both cases make the piston decidedly heavy. Where lightness is desired, as is often the case with a conical piston, when it is a steel casting, the thickness of the piston near the flange, as at t, Fig. 18 (a), may be made from $.0015\,D\sqrt{p}$ in large engines to $.003\,D\sqrt{p}$ in small engines, but should never be less than .625 inch, p being the maximum unbalanced steam pressure on the piston in pounds per square inch. The thickness of the body of the

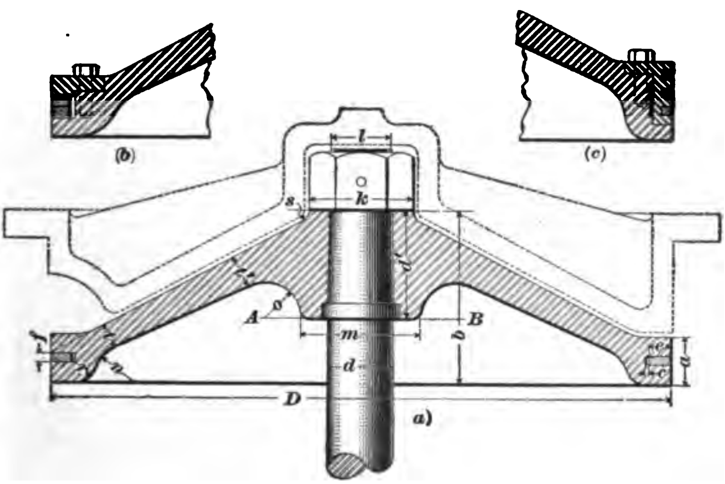


FIG. 18

piston near the boss, as t', may be made 1.75 times the thickness of the body near the flange.

49. If desired, the stress per square inch on the body of the piston near the hub, which will be the maximum stress existing in the piston, may be calculated by the formula:

$$S = \frac{p \operatorname{cosec}^2 a\,(2\,D^3 - 3\,k\,D^2 + k^3)}{12\,t'\,(D - k)^2 \operatorname{cosec} a \cot a + 4\,t'^2\,(k - t' \cos a)},$$

in which D = diameter of cylinder, in inches;
S = stress in material, in pounds per square inch;
k = diameter of piston boss;

p = maximum unbalanced steam pressure on piston, in pounds per square inch, which should not be less than 113;
a = angle between outside of conical body of piston and axis of piston rod;
t' = thickness of body of piston near the boss.

If S, as calculated by the foregoing formula, has an excessive value, the dimensions of the piston should be modified.

In Fig. 18 (*a*) and (*b*) are shown methods of attaching the packing rings to large pistons so that the rings may be removed for inspection or repairs without taking the piston out of the cylinder. This is a very important advantage in many cases, especially for marine work, where the pistons are often very heavy and facilities for handling them are poor.

PISTON PACKING

50. It is, of course, impossible to turn the piston to exactly fit the cylinder at all temperatures; therefore, the piston is made slightly smaller than the cylinder bore, and some form of packing is used to prevent the steam from leaking through between the piston and cylinder walls.

The simplest and about the best form of packing, particularly for small pistons, is the cast-iron ring shown in cross-section at s, s, Figs. 13 and 15. These rings are generally of uniform thickness. Many makers, however, prefer to make the thickness where the rings are cut about half the thickness at the opposite side.

The proportions used for the spring packing ring shown in Figs. 13 and 15 are as follows:

Thickness and depth of rings the same and equal to

$$.135\sqrt{2D} - .14 \text{ inch},$$

where D is the diameter of the cylinder.

In Fig. 19 (*a*) and (*c*), the packings for the pistons shown in Figs. 17 and 18 are shown in detail. In addition to the dimensions given in connection with the pistons, the following proportions are to be used:

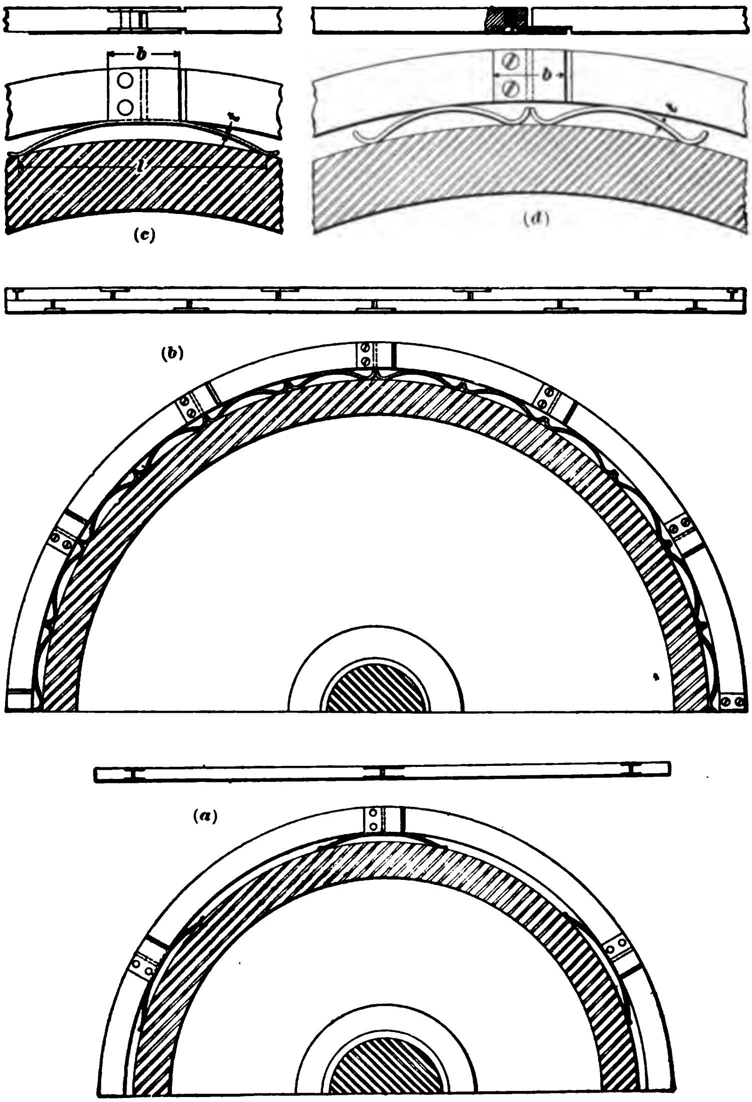


FIG. 19

b = 3 inches.
l = 9 inches.
t = .0625 inch.

Details of the packing shown in section in Fig. 18 (b) are shown in Fig. 19 (b) and (d). The proportions applying here are:

b = 3 inches.
t = .09375 inch.

The length of the segments should be about 15 inches; two springs are placed behind each segment. The packing rings shown in Fig. 19 are usually made of cast iron and the springs of steel.

51. Fig. 20 shows **Tripp's patent piston packing.** The rings s, s are made of cast iron, being split so as to

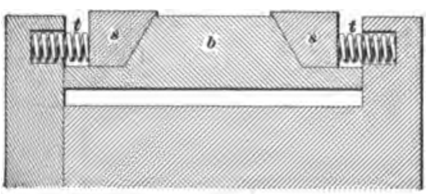


FIG. 20

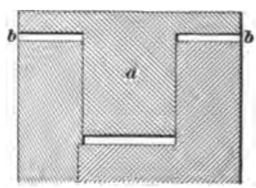


FIG. 21

spring outwards against the cylinder walls. They are supported by an adjustable ring b made with conical surfaces against which the packing rings bear. The pressure of the steam against the packing ring forces it against this conical surface, thus tending to open out the ring and make it press against the cylinder. The spiral springs t, t are for the purpose of holding the packing rings in place when they are not acted on by steam pressure.

Fig. 21 shows a style of ring packing much used for piston valves. At a is a split or sectional cast-iron ring, which is forced out against the walls of the cylindrical valve seat by the pressure of the steam in the spaces b, b between the overhanging parts of the ring and the main part of the piston or valve.

PISTON ROD

52. Piston-Rod Calculations.—The piston rod of the ordinary double-acting steam engine is subjected to alternate tension and compression of substantially equal magnitude. With the usual proportions of length to diameter, the rod should be treated as a long column, or strut, under the action of compression. As a column under compression, the rod should be considered as fixed transversely at one end and free at the other; this is to be distinguished from a column fixed at one end and guided at the other, which is eight times as strong. The formula used for such cases is based on Euler's column formula. The diameter of the piston rod is therefore given by the formula:

$$d = 2\sqrt[4]{\frac{4 f p D^2 L^2}{10 E}}, \qquad (1)$$

in which d = diameter of piston rod, in inches;

f = factor of safety, which in low-speed engines has an average value of about 10, and in high-speed engines, an average value of about 20, in good practice;

p = maximum unbalanced steam pressure on piston, in pounds per square inch, taken not less than 113;

D = diameter of cylinder, in inches;

L = length of piston rod from piston to crosshead, in inches;

E = modulus of elasticity of material, which may be taken as 25,000,000 for wrought iron and 30,000,000 for steel.

With a given material and a given steam pressure, such that E, f, and p may be taken as constant, formula **1** reduces to the form

$$d = K\sqrt{D L}, \qquad (2)$$

in which K represents $2\sqrt[4]{\frac{4 f p}{10 E}}$.

An examination of a large number of steel piston rods stationary engines designed for a steam pressure of 100 pounds, gauge, shows the following values of K: In good practice, for high-speed engines, K varies from .119 to .177, with an average value of .145, and for low-speed engines, K varies from .098 to .136, with an average value of .112.

Formula **1** is rational, and may be used in all cases, while formula **2** is more convenient, and may be used directly in calculations for ordinary stationary engines. For other classes of work, formula **2** should not be used, except to check the results obtained by the first formula.

EXAMPLE.—Calculate the diameter of a steel piston rod 21 inches long, the diameter of the cylinder being 13 inches, the steam pressure 90 pounds, and the speed 120 revolutions per minute.

SOLUTION.—Though the expected steam pressure is only 90 lb., the rod should be calculated for a maximum unbalanced pressure of 113 lb. per sq. in. As the speed is 120, the engine is low-speed, and f may be taken as 10. Then, substituting in formula **1**,

$$d = 2\sqrt[4]{\frac{4 \times 10 \times 113 \times 13 \times 13 \times 21 \times 21}{10 \times 30{,}000{,}000}} = 2.059, \text{ say } 2\tfrac{1}{16}, \text{ in.}$$

Check the result by using formula **2**, with $K = .112$. Then, by substituting,

$$d = .112\sqrt{13 \times 21} = 1.85, \text{ say } 1\tfrac{7}{8}, \text{ in}$$

As this result is somewhat smaller than the value found by formula **1**, the value of K might be taken as .136 and again substituted in formula **1**. This gives

$$d = .136\sqrt{13 \times 21} = 2.247, \text{ say } 2\tfrac{1}{4}, \text{ in.}$$

Hence, the solution by formula **1** for low-speed stationary engines is satisfactory, and the value of d thus found may be considered as justified by practice and may be accepted. Ans.

53. Connection of Rod to Piston.—Modes of fastening the piston rod to the piston are shown in Figs. 13 to 18. The end of the rod is tapered and threaded to receive a nut or it is riveted. The taper may vary in different cases from $\frac{1}{2}$ to 3 inches per foot. The cross-section of the rod at root of threads should be such as to give a tensile strength of 5,000 pounds per square inch for wrought iron and 7,000 pounds for steel. Letting d_1 be the diameter of rod at root of thread,

$$.7854\, d_1^2 \times 5{,}000 = .7854\, D^2 p \text{ for wrought iron}$$

and

$$.7854\, d_1^2 \times 7{,}000 = .7854\, D^2 p \text{ for steel}$$

Or, $d_1 = .014\, D \sqrt{p}$ for wrought iron

and $d_1 = .012\, D \sqrt{p}$ for steel

Rods having collars forged on to bear against the piston are much to be preferred for heavy pressures. When the section of a rod is to be reduced, it is important, especially for a steel rod, to provide a liberal fillet, as shown in Fig. 14, so as to leave no sharp corners.

CONNECTING-ROD

CALCULATION FOR CONNECTING-ROD

54. The first dimension of the connecting-rod that should be determined is the length. A very short rod is objectionable, because the shorter the rod the greater is the thrust along it, due to the steam pressure on the piston, and the greater are the irregularities introduced into the steam distribution by the angularity of the connecting-rod. On the other hand, increasing the length of the connecting-rod increases the stresses in it, as a column in compression, so that an unnecessarily long rod should also be avoided.

As a working compromise, in stationary practice, the length of connecting-rod between centers of crankpin and wristpin is generally made between 5.5 and 6 times the length of the crank-arm; it is seldom that the connecting-rod of a stationary engine will go beyond these limits. However, if it is necessary to shorten the engine, the length of connecting-rod may be considerably less. In naval engines, the ratio of the connecting-rod to the crank is often as small as 4, and sometimes only 3.5.

The connecting-rod is subjected to alternate tension and compression; therefore, in design it is to be treated as a long column under compression. Considered as a column, the connecting-rod is pin-ended and guided at both ends in the plane of motion and is square-ended in the plane at right angles.

55. In low-speed engines, the connecting-rod frequently has a circular cross-section; and, in the older practice at least, was usually swelled from the necks near the stub ends to a maximum diameter at the middle. With a rod of this kind, the diameter at the middle, which will be the section of maximum stress, is given by the following formula, which is based on Euler's formula:

$$d = 2\sqrt[4]{\frac{f p D^2 L^2}{10 E}}, \qquad (1)$$

in which d = diameter of rod, in inches;

f = factor of safety, which for this kind of rod varies in good practice from about 8 to about 23, the average value being about 16;

p = maximum unbalanced steam pressure on piston, in pounds per square inch, and should not be less than 113;

D = diameter of cylinder, in inches;

L = length of connecting-rod from center of wrist-pin to center of crankpin, in inches;

E = modulus of elasticity of material, which for steel is 30,000,000.

With a given material and a definite steam pressure, the values of E, p, and f may be taken as constant, and formula **1** reduces to

$$d = K\sqrt{D L}, \qquad (2)$$

in which the value of K is $2\sqrt[4]{\frac{f p}{10 E}}$.

An examination of steel connecting-rods of this kind designed for low-speed stationary engines carrying a steam pressure of 100 pounds, gauge, shows that in good practice K has a value from .0816 to .105, with an average value of .0935.

Formula **1** may be used in the design of all connecting-rods of circular cross-section. Formula **2** may be used directly in the design of the class of rods for which the constant K is derived; but, for other rods, it should not be used except as a check on the results given by formula **1.**

56. The usual custom is to taper the rod, and if it is tapered both ways the diameter at the middle should be determined by formula **1**, Art. **55.** The diameters at the necks, next to the stub ends, are then usually calculated for simple tension and compression by the formula

$$d' = D\sqrt{\frac{p f}{S_c}},$$

in which d' = diameter at neck, in inches;

S_c = maximum compressive stress in material, in pounds per square inch, which for mild steel may be taken as 50,000.

The other symbols are the same as for formula **1**, Art. **55.** If, in any particular case, a value has been assigned to f, to be used in formula **1**, Art. **55**, the same value should be assigned to f in the formula given in this article. The rod is then given a uniform taper both ways, from middle to neck, and the corners are filleted or rounded off as required.

If the rod tapers from crosshead to crank end, which is preferable to having it taper both ways, except for low speeds, the diameter at the crosshead neck should be determined by the formula of this article. The diameter of the middle should be determined by formula **1**, Art. **55**, and the rod should then be given a uniform taper throughout.

57. In high-speed engines, the rod is often rectangular or approximately so in cross-section, and is usually deepest at the crank end. In this class of rods, the breadth at the middle, that is, the dimension perpendicular to the plane of motion of the rod, may be determined by the formula

$$b = .588\sqrt[4]{\frac{f p}{E}}\sqrt{D L}, \qquad (1)$$

in which b = breadth of rod at middle, in inches;

f = factor of safety, which in good practice varies from about 9 to about 57, the average value being about 20.

The other symbols are the same as for formula **1**, Art. **55.**

Where f, p, and E may be taken as having definite values, formula **1** reduces to

$$b = K\sqrt{D\,L}, \qquad (2)$$

in which K represents $.588\sqrt{\frac{f\,p}{E}}$.

An examination of steel connecting-rods of this class in a large number of high-speed stationary engines carrying steam pressures of 100 pounds, gauge, shows that K varies from .0433 to .0693, with an average value of .0545 in good practice.

In rods of this class, the breadth at the mid-section having been determined, the height at the mid-section, that is, the dimension parallel with the plane of motion, may be taken as from 2.18 to 4 times the breadth, the average ratio of height to breadth at mid-section being 2.75. The greater the piston speed and rotative speed, the greater should be the ratio of height to breadth at this section.

Formula **1** may be used in the design of all connecting-rods of rectangular cross-section. Formula **2** should be used directly only in the design of such rods as those from which the value of K is derived, and should not be used in other cases, except as a check on the results obtained by formula **1**.

58. While the formulas in Art. **57** give results agreeing quite closely with good practice, some investigators think it advisable to check the unit stress in a rod calculated by their use. This is done by means of the following formula, which is known as **Ritter's formula.** This check is suggested owing to the wide range of the factor of safety and the consequent danger of passing the limit of safety in the use of Euler's formula,

$$S = \frac{P}{A}\left(1 + \frac{S_e}{\pi^2\,m\,E} \times \frac{l^2}{r^2}\right),$$

in which S = greatest unit compressive stress in rod, in pounds;

P = maximum thrust on connecting-rod, in pounds;

A = area of cross-section of rod, in square inches;

S_e = unit stress of the material at elastic limit; usually taken as 26,000 for wrought iron and 35,000 for steel;
π^2 = 9.8696;
m = a constant = 4 for plane at right angles to plane of motion of rod;
E = modulus of elasticity, which is 25,000,000 for wrought iron and 30,000,000 for steel;
l = length of column, being taken in this case from center of crankpin to center of wristpin;
r = radius of gyration, being $\frac{1}{16}d^2$ for circular cross-sections of rod and $\frac{1}{12}a^2$ for rectangular cross-sections;
d = diameter of circular rod, in inches;
a = least dimension of rectangular rod, in inches.

The value of S found by the formula just given should not exceed the elastic limit as given for S_e.

59. The crosshead neck should be designed for simple tension and compression. The breadth should be retained as determined for the mid-section, and the height should be determined by the formula

$$h' = \frac{.7854\,D^2 p f}{b\,S_c},$$

in which h' = height of rod in crosshead end neck, in inches;
D = diameter of steam cylinder, in inches;
p = maximum unbalanced steam pressure on piston, in pounds per square inch, which should not be less than 113;
f = factor of safety, which, in any particular case, should be taken the same as is used in calculating the mid-section by formula **1**, Art. **57**;
b = breadth of rod in crosshead end neck, in inches;
S_c = maximum compressive stress in material, in pounds per square inch, which, for mild steel, may be taken as 50,000.

After the dimensions of these two cross-sections are determined, the rod is usually made of equal thickness throughout, and is given a uniform taper from the neck at the crosshead end to the crankpin end.

PROPORTIONS FOR CONNECTING-RODS

60. Solid and Open Connecting-Rod Ends.—The design of the ends of the rod is generally based on empirical expressions deduced from practice. The proportions that follow are based on the practice of the best engine builders. Fig. 22 shows a style of rod that gives excellent results in stationary work. The crosshead end is forged solid and cut out for the brasses, which are made without top and bottom flanges on one side, so that they can be slipped into the rod. The brasses are held in place and adjusted by the steel wedge w_1 and the adjusting screws S_1. The brasses on the crosshead ends of connecting-rods are seldom babbitted, as experience shows that it is unnecessary.

The crank end of the rod is made fork-shaped, so that the brasses can be put on the crankpin and then slipped into the rod from the end. If the wristpin cannot be removed from the crosshead, such a construction must also be used for the crosshead end of the rod. The bolt B, which is turned and fitted in a reamed hole, holds the brasses, which are adjusted by a steel wedge and screws, in the same manner as for the crosshead end. The crankpin brasses are babbitted, as shown.

61. The dimensions of this rod are to be made according to the following proportions, which are suitable for a rod of either steel or wrought iron.

For the wristpin, or crosshead pin, end:

D = diameter of cylinder, in inches.

d = diameter of crosshead pin, in inches, which will be treated in *Steam-Engine Design*, Part 2.

$a = 1.42d$.

$b = 1.125d + .375$ inch.

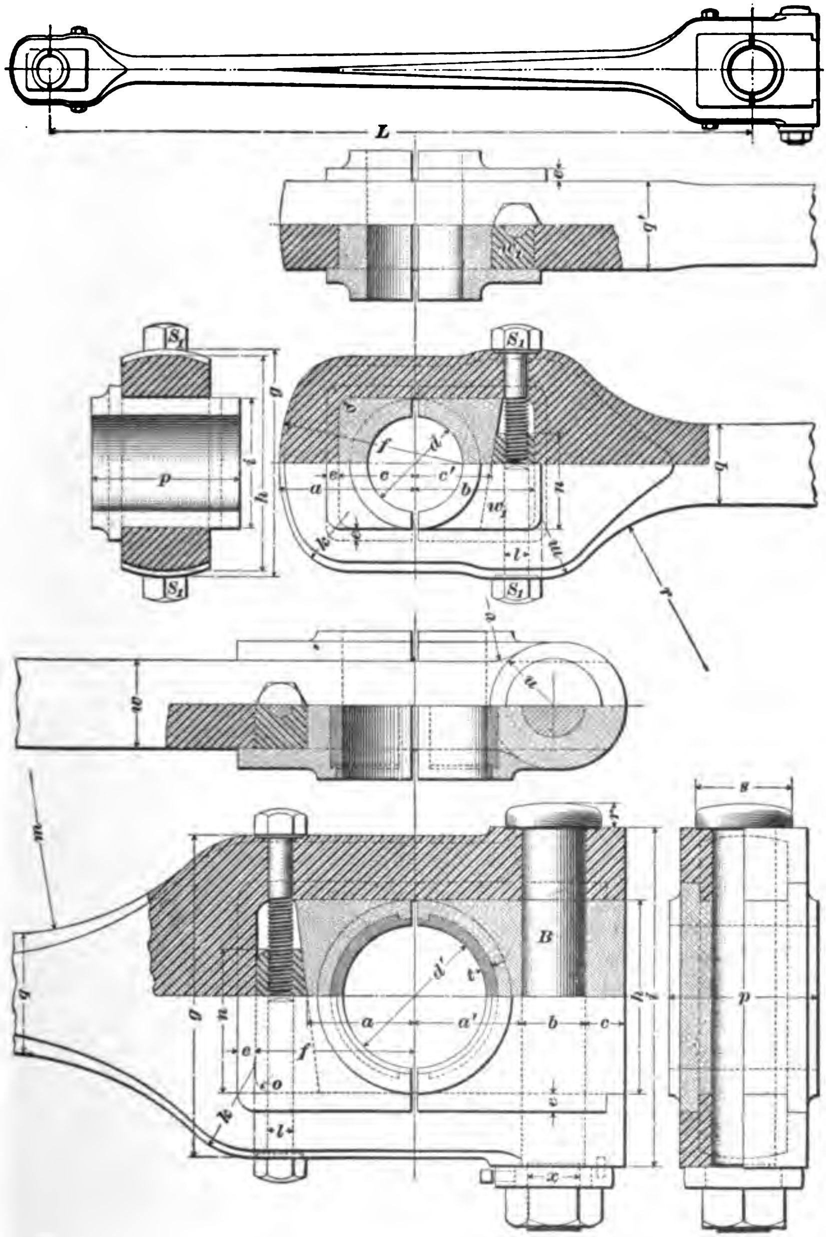


FIG. 22

$c = .75\,d + .125$ inch.
$c' = .75\,d + .125$ inch.
$e = .125\,d$.
$f = 1.92\,d$.
$g = 2.375\,d$.
$h = 2.25\,d$.
$i = 1.35\,d$.
$k = .625\,d$.
$l = .035\,D + .25$ inch.
$m = .625\,d$.
$n = d$.
$o = .125\,d$.

p = length of wristpin journal − .125 inch in small engines, and length of wristpin journal − .25 inch in large engines.
q should be designed by the formula of Art. **56**, q being the same as d' in that formula.
$q' = 1.1\,q$.
$r = 1.75\,d$.

Taper of adjusting wedges = 1¼ inches per foot.

For the crankpin end:

D = diameter of cylinder, in inches.
d' should be designed by the rules for finding the diameter of crankpin, according to Arts. **34** to **37**.
$a = .75\,d'$.
$a' = .75\,d'$.
$b = .1\,D + .4375$ inch.
$c = .625\,b$.
$e = .125\,d'$.
$f = d' + .5$ inch.
$g = 2.25\,d'$.
$h = 1.35\,d'$.
$i = 2.375\,d'$.
$k = .625\,d'$.
$l = .035\,D + .25$ inch.
$m = 2\,d'$.
$n = d'$.
$o = .125\,d'$.

p = length of crankpin journal − .125 inch in small engines, and length of crankpin journal − .25 inch in large engines.
q should be determined by the formulas given for designing the crankpin neck of the connecting-rod in Art. **56**.
$r = .375\,b$.
$s = 1.5\,b$.
$t = .02\,D + .0625$ inch.
$u = b$.
$v = b$.
w should be designed by formula **1**, Art. **57**, w being the same as b of that formula.
$x = .8\,b$.

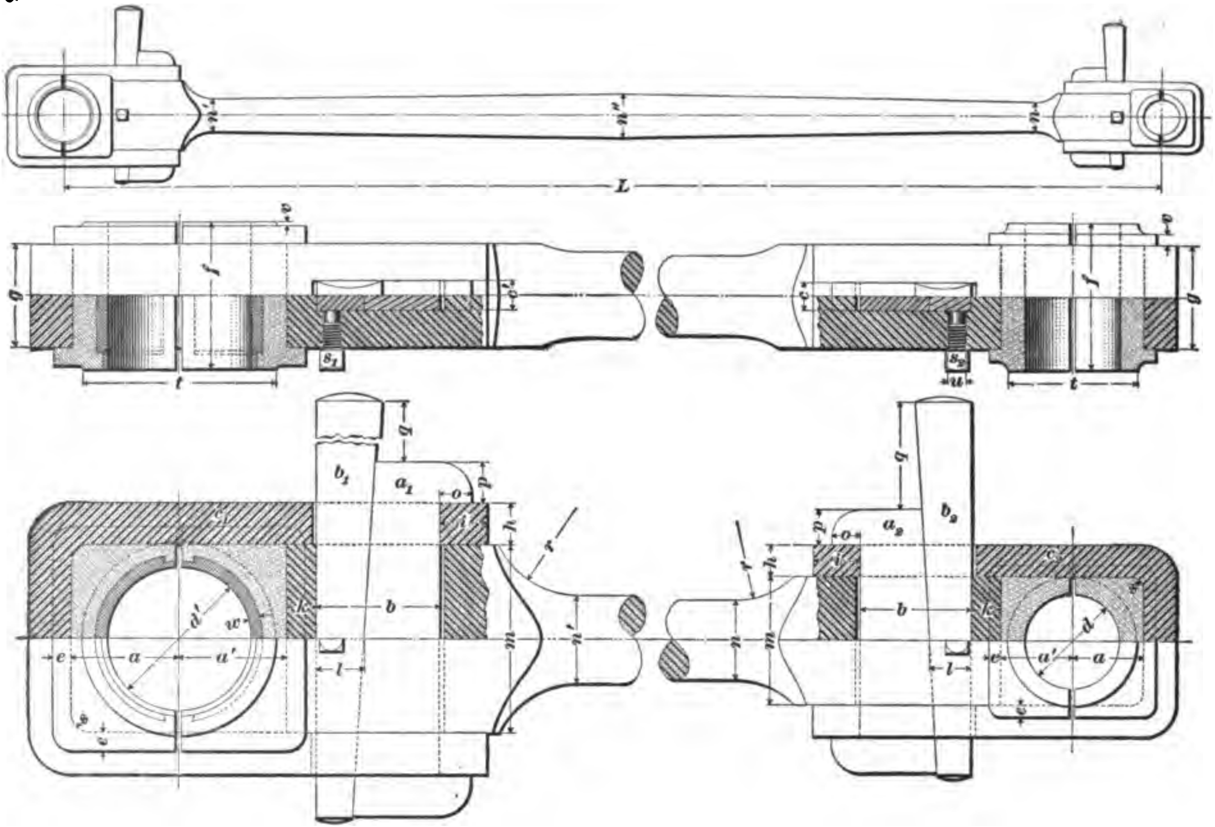


FIG. 23

In designing the cross-section of the rod at the middle of its length, the breadth should be determined by formula **1**, Art. **57**. At the point where the breadth is measured, the corners of the cross-section should not be rounded off enough to reduce the height to less than twice the breadth. The total height should be from 2.18 to 4 times the breadth, the average being 2.75.

The nut and locking collar should be designed according to the principles of machine design.

The taper of the adjusting wedge may be made 1½ inches per foot.

62. Strap-End Connecting-Rod.—In Fig. 23 is shown a **strap-end connecting-rod.** This rod is commonly used and has been found very satisfactory. However, the strap end should not be used on the crankpin, where either the piston speed or the rotative speed is high. The straps c_1 and c_2 are fastened to the ends of the rods by means of gibs a_1 and a_2 and cotters b_1 and b_2. The cotters are held in place by setscrews s_1 and s_2. Small steel blocks, shown between the ends of the setscrews and the cotters, are used to prevent the cotter from being injured by the setscrews.

The rod, cotters, gibs, and straps are generally made of steel, but wrought iron is occasionally employed. The crankpin brasses are shown babbitted, and the wristpin brasses without Babbitt. The brasses are adjusted by means of the cotters, which draw the straps farther on to the rod when they are driven in.

The dimensions for the rod shown in Fig. 23 are given by the following proportions:

For the wristpin end:

D = diameter of cylinder, in inches.

d = diameter of wristpin, in inches, which will be treated in *Steam-Engine Design*, Part 2.

$x = .7854\,n^2$ = a factor for use in finding proportions that follow.

$a = .75\,d + .125$ inch.

$a' = .75\,d + .125$ inch.

$b = \sqrt{2.5\,x}$.

$c = .25\,b$.

$e = .125\,d$.

f = length of wristpin journal $-$ .125 inch in small engines, and length of wristpin journal $-$ .25 inch in large engines.

$g = 1.3\,n$.

$h = \dfrac{.5\,x}{g - c}$.

$i = \dfrac{.32\,x}{h}$.

$k = \dfrac{x}{1.8\,d}$.

$l = .375\,b$.

$m = 1.35\,d$ for wristpins up to 3.5 inches in diameter.

$m = 1.48\,n$ for pins above 3.5 inches in diameter.

n should be determined by the formula of Art. **56**, being the same as d' of that formula.

$o = .25\,b$.

$p = .33\,b$.

$q = 1.125\,d$ for wristpins up to 3.5 inches in diameter, and $q = 4$ inches, constant, for pins above 3.5 inches in diameter.

$r = n$.

$s = .125\,d$.

$t = 1.35\,d$.

$u = .02\,D + .25$ inch.

$v = .125\,d$.

The taper of the cotter is $\frac{3}{4}$ inch per foot.

For the crankpin end:

D = diameter of cylinder, in inches.

d' should be determined by the rules for finding the diameter of crankpins, given in Arts. **34** to **37**.

n' should be found by the formula of Art. **56**, being the same as d' of that formula. In practice, the value of d' found by the formula is often multiplied by 1.1, to find the value of n', Fig. 23. Though, theoretically, this increase in the diameter of the neck is unnecessary, it may be considered good practice as an empirical allowance against unknown stresses due to inertia and seizing of the crankpin brasses from insufficient lubrication.

$x' = .7854\,n'^2$ = a factor used in the proportions that follow.

$a = .75\,d'$.

$a' = .75\,d'$.

$b = \sqrt{2.5\,x'}$.

$c' = .25\,b$.

$e = .125\,d'$.

f = length of crankpin journal − .125 inch, in small engines, and length of crankpin journal − .25 inch in large engines.

$g = 1.3\,n$, the same as for wristpin end.

$h = \dfrac{.5\,x'}{g - c'}$.

$i = \dfrac{.32\,x'}{h}$.

$k = \dfrac{x}{1.8\,d'}$, the same as for wristpin end.

Taper of cotter = $\frac{3}{4}$ inch per foot.

$l = .375\,b$.

$m = 1.3\,d'$.

$o = .25\,b$.

$p = .33\,b$.

q = same values as for wristpin end.

$r = 1.1\,n$.

$s = .25\,d'$.

$t = 1.35\,d'$.

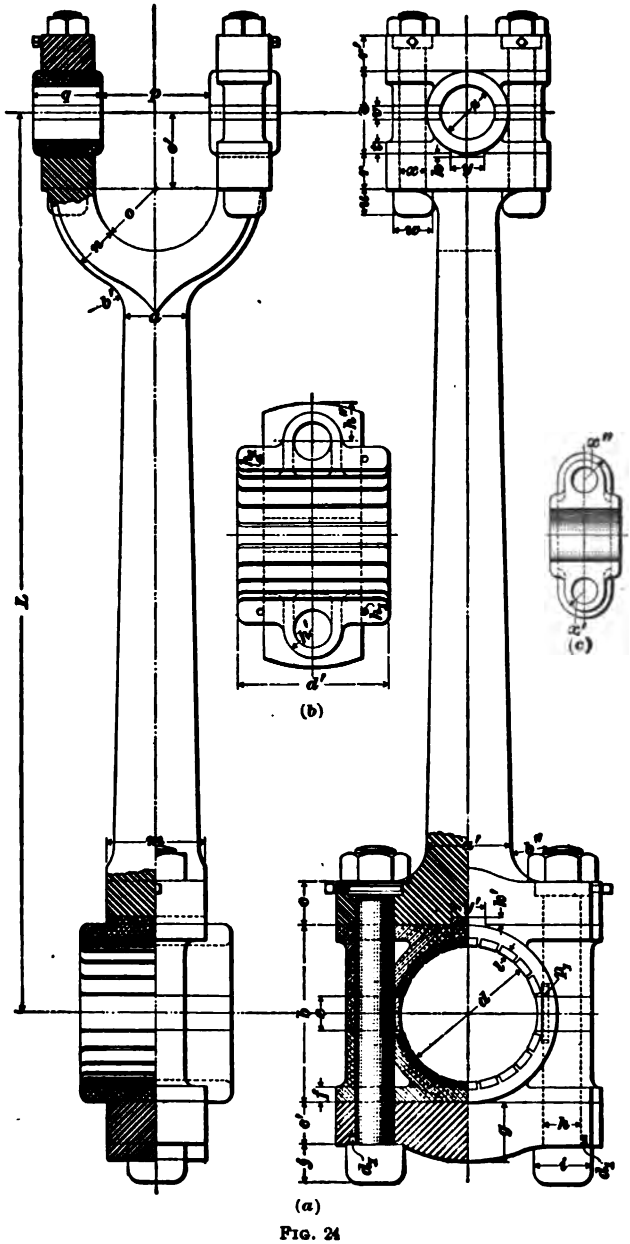
v = .125 inch (constant).

$w = .02\,D + .0625$ inch.

n'' should be found by formula **1**, Art. **55**, being the same as d of that formula.

63. Marine-End Connecting-Rod.—Fig. 24 shows a connecting-rod with **marine ends.** For marine engines, and in some cases for stationary engines, the crosshead end is forked as shown. For most stationary work, however, the crosshead end is not forked, and a solid or strap end is often used with a marine crankpin end. The object in using the forked end is to obtain the most solid and substantial form of crosshead. Where this is not necessary, the forked-end rod should be avoided, because it is impossible to adjust the wristpin brasses so accurately as to divide the pressure equally between them. The inequality of pressure on the wristpins brings a side thrust on the crosshead and thus increases the stresses in the connecting-rod.

In the connecting-rod shown in Fig. 24 (a), the brasses are held to the ends of the rod, which are forged to a **T** shape, by turned bolts. These bolts pass through steel or wrought-iron caps and the brasses. Liners are fitted between the two parts of each set of brasses, and when the brasses become worn, the liners are taken out and either filed or planed down, thus allowing the brasses to be tightened.



(a)

Fig. 24

The crankpin brasses are babbitted, the Babbitt being poured so as to project about $\frac{1}{8}$ inch above the brass in order that the pin will bear only on the Babbitt, with oilways between the Babbitt blocks.

The liners in the crankpin end are sometimes secured by small pins p_1, which pass through the liner and project into the holes h_1, shown in the view of the brass given in Fig. 24 (*b*). Fig. 24 (*c*) shows a view of one of the wristpin brasses. The rod and caps are usually made of steel, although wrought iron may be used.

64. The dimensions for the rod shown in Fig. 24 (*a*) are determined by the following proportions:

a = diameter of rod at wristpin end, and should be calculated by the formula of Art. **56**, a being the same as d' of that formula. In marine work, good material and workmanship should be employed, so that a small factor of safety can be used in design, as it is imperative to reduce weight.

a' = diameter of rod at crankpin end, and should be determined according to Art. **56**.

$b = 1.3\,d$.

$b' = .6\,a$.

$b'' = .5\,a'$.

$c = .7\,a$.

$c' = .7\,a$.

d = diameter of crankpin.

d' = length of crankpin journal − .125 inch.

$e = .25\,d$.

$f = .35\,c$.

$g = 1.4\,c$.

h = diameter of bolt at root of thread, when area of cross-section at root of thread is $.00008\,A\,P$ for steel and $.0001\,A\,P$ for wrought iron, where A is the area of cylinder, in square inches, and P the boiler pressure, in pounds per square inch, which should not be less than 113.

$h' = .875\,h$.

$h'' = h$.

$i = 1.5\,h.$

$j = h.$

$k = .05\,z + .0625$ inch.

$k' = l.$

$l = .05\,d + .0625$ inch.

$m = 1.1\,a'.$

$n = .7\,a.$

$o = .5\,p + .5\,(q - 2\,x).$

$o' = .6\,a + .675\,z.$

$p = 2\,B + .25$ inch, where B is the diameter of piston rod.

q = length of wristpin as made for crosshead.

$r = .6\,a.$

$r' = .6\,a.$

$s = 1.68\,z.$

$t = .35\,r.$

$u = x.$

$v = .25\,z.$

$w = 1.5\,x.$

x = diameter of bolt at root of thread, when area of cross-section is $.00004\,AP$ for steel and $.00005\,AP$ for wrought iron.

$y = .25\,d.$

$y' = .25\,d.$

z = diameter of wristpin as made for a solid crosshead.

$x' = .875\,x.$

$x'' = x.$

Proportions for locknuts should be determined according to the principles of machine design. The dowel-pins shown at *d*, Fig. 24 (*a*), are used to keep the bolt from turning with the nut. The distance between the bolts should be such that the brass will be about .25 inch thick at the thinnest part between the bolt and the pin.

65. In marine design, all the connecting-rods for the main engines should be made alike, so that they may be interchangeable. This is done so that a large quantity of duplicate parts will not have to be carried. In the case of a marine engine with several cylinders, the designs should be worked out for all the cylinders, and the heaviest design found necessary for any cylinder should be adopted for all. The same rule applies to all marine-engine parts that can be made interchangeable.

STEAM-ENGINE DESIGN

(PART 2)

ENGINE DETAILS

CROSSHEADS

1. Corliss Engine Crosshead.—**Crossheads** are made in a great variety of forms, some of the most important of which, together with the proportions for their design, are given in the following pages. Fig. 1 shows a crosshead much used on Corliss and similar engines. This crosshead is composed of a cast-iron box, with a boss for a piston rod and bosses for the wristpin. The wristpin is generally made of steel—occasionally of wrought iron—with the ends tapered so as to fit snugly into the crosshead. This pin is held by a nut and washer and has a projecting part y, which is drilled for an oil cup. Holes are drilled into the pin, as shown by the dotted lines, for the purpose of leading oil to the connecting-rod brasses.

2. Shoes or gibs g_1 are fitted to the crosshead on tapered ways, and these gibs can be adjusted by means of the bolts t_1. The gibs shown on the crosshead have cylindrical bearing surfaces, but the cross-section of a gib with the bearing surface V-shaped is shown at A. The engine guides are made either cylindrical or V-shaped, to correspond with the gibs.

Let s = diameter of wristpin, in inches;

t = length of wristpin journal, in inches.

Then, s and t are given by the following formulas:

$$s = \sqrt[3]{\frac{t D^2 p}{S_4}}, \qquad (1)$$

$$s t = \frac{.7854 D^2 p}{P}; \qquad (2)$$

in which D = diameter of cylinder, in inches;

p = maximum unbalanced pressure on the piston, in pounds per square inch;

S_4 = safe bending stress in the material of the pin, in pounds per square inch; as the crosshead pin, or wristpin, works under a rapidly reversing load and is also subject to considerable shock, S_4 should not exceed about 7,000, except in case of unusually good material and workmanship and careful inspection, when it may equal 8,000;

P = maximum allowable pressure on the bearing, in pounds per square inch of projected area. P may have a value from 800 to 1,400, but the best practice gives 1,200 to 1,350 as the highest values, the lower values of P corresponding to the higher rotative speeds.

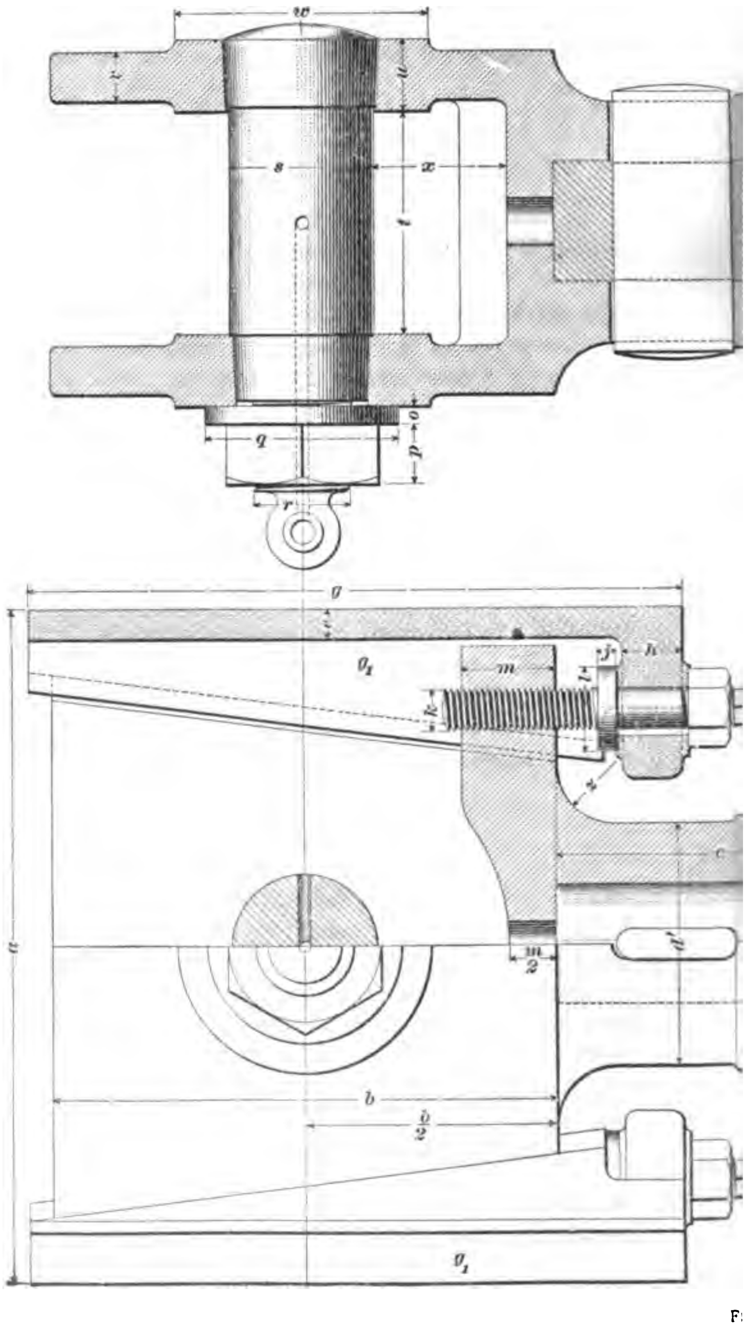
If p and P have definite values, formula **2** may be written

$$s t = K D^2, \qquad (3)$$

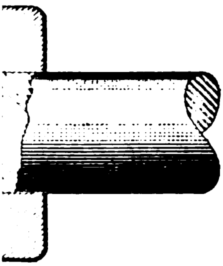
in which K is a constant representing $\frac{.7854 p}{P}$.

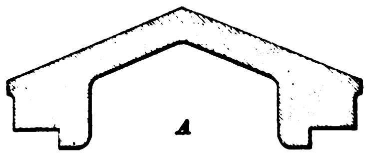
An examination of the wristpins of a large number of low-speed, side-crank stationary engines, in which the type of crosshead shown in Fig. 1 is generally used, shows K to have a value ranging from .042 to .083, with an average value of .058 in good practice. The value of t in terms of s may be found by formula **2**. By substituting this value of t in formula **1**, s may be found, after which t may be definitely determined by substituting the value of s in formula **2**. Formulas **1** and **2** apply to all wristpins supported at both ends.

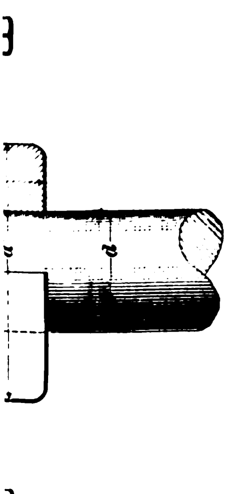
3. Let f represent the width of the shoe, in inches, as shown in Fig. 1, and g the length of the shoe, in inches.

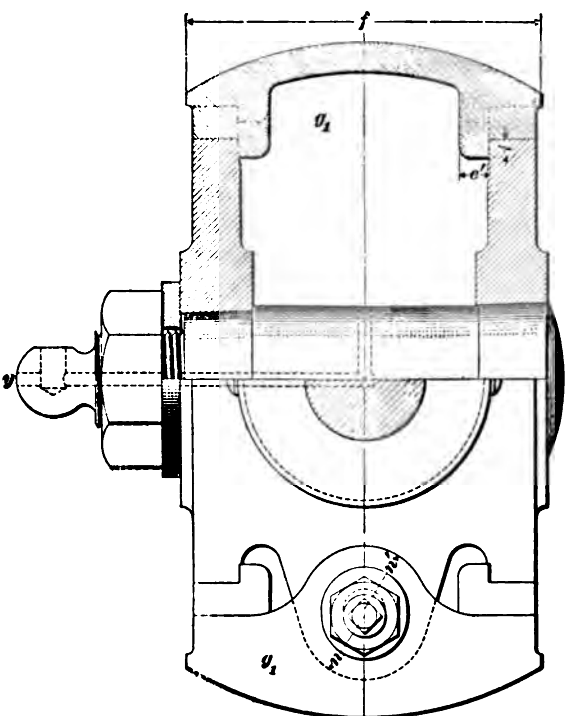


Fi




A


u
d


f
g1
e'
v
g1

Then, for cylinders up to and including 26 inches in diameter,

$$f = .357\,D + 1.625 \text{ inches}$$

For cylinders larger than 26 inches in diameter,

$$f = .376\,D + 1 \text{ inch}$$

The length of the shoe g is given by the formula

$$g = .7854\frac{D^2 p}{f P\sqrt{n^2 - 1}}, \qquad (1)$$

in which P = maximum allowable pressure on the guide, in pounds per square inch;

n = ratio of length of connecting-rod between center of crankpin and center of wristpin to length of crank between center of crankpin and center of shaft;

D = diameter of cylinder, in inches;

p = maximum unbalanced steam pressure, in pounds per square inch, which should not be taken less than 113.

An examination of a large number of engines shows the average value of P to be 36 for low-speed stationary engines and 27 for high-speed engines. Some authorities give values varying from 40 to 100, but the values just stated represent the best American practice, and as these values are safer, they are to be preferred.

In low-speed stationary engines, the value of n is about constant. Then, in this class of engines, with definite values for p and P, formula **1** reduces to

$$fg = KD^2 \qquad (2)$$

An examination of low-speed stationary engines shows the value of K to vary from .23 to .52, with a value of .37 as the average in good practice. The higher the piston speed, the larger the value that should be assigned to fg.

4. The remaining dimensions of the crosshead shown in Fig. 1, are given by the following proportions:

D = diameter of cylinder.

d = diameter of piston rod.

a must be found by making a scale drawing of the connecting-rod in its extreme position. Care must be taken that the rod will clear the gibs, and also the guides, for all positions.

b = length of crosshead body, and must be given such a length that $\frac{b}{2}$ will accommodate the end of the connecting-rod to be used. This length can best be determined on a drawing board.

$c = 2.125\,d$.

$d' = 2\,d$.

$d'' = 2.125\,d$.

$e = .01\,D + .5$ inch.

$e' = e$.

$h = .04\,D + .5625$ inch.

$i = .6\,e$.

$j = .015\,D + .125$ inch.

$k = .04\,D + .3125$ inch.

$l = .08\,D + .625$ inch.

$m = .12\,D + .125$ inch.

$n = .06\,D + .5$ inch.

$n' = n$.

$o = .19\,r$.

$p = .625\,r$.

$q = 2.125\,r$.

$r = .66\,s$.

s = diameter of wristpin, in inches (see Art. 2).

t = length of wristpin journal, in inches (see Art. 2).

$u = .05\,D + .625$ inch.

$v = .043\,D + .3125$ inch.

$w = 1.75\,s$.

x must be determined by making a scale drawing of the wristpin end of the connecting-rod, and must be such that the connecting-rod will swing clear of the crosshead in all positions by at least $\frac{1}{8}$ inch.

y is plug for oil hole.

$z = .5\,D$.

Taper of gibs, 1.5 inches per foot.

For cylinders above 20 inches in diameter, the gibs should be ribbed.

This crosshead is designed to be used with a solid-end connecting-rod.

5. Marine-Engine Crosshead.—Fig. 2 shows a style of crosshead used mostly for marine work. The two wristpins a_1, a_1 and the block b_1 are one solid steel forging. This crosshead requires a forked-end connecting-rod. The bearing surfaces are composed of two cast-iron shoes fastened to the block by bolts. These shoes are babbitted, the Babbitt

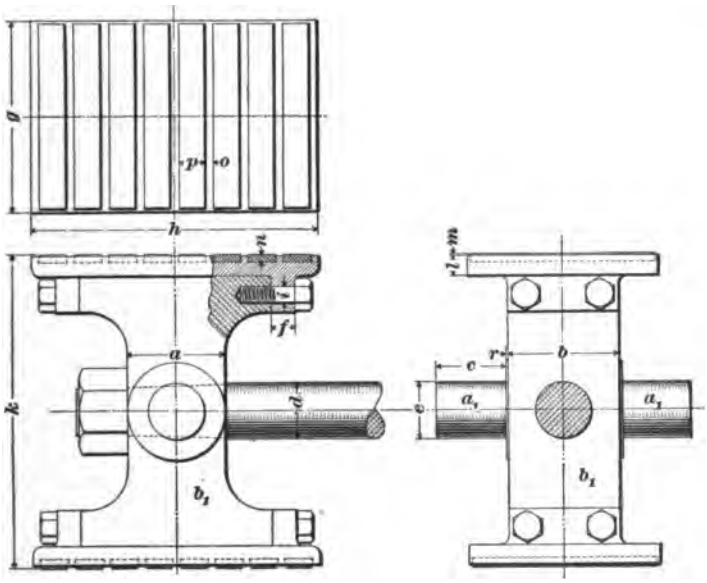


Fig. 2

being dovetailed into and raised a little above the surface of the iron, so that no wear will come directly on the iron. The piston rod is tapered in the block and fastened with a nut.

Let e = diameter of wristpins, as shown in Fig. 2;

c = length of each wristpin.

Then, e and c are given by the following formulas:

$$e = \sqrt[3]{\frac{2.67\ D^2 p c}{S_4}}, \qquad (1)$$

$$ec = \frac{.524\ D^2 p}{P}; \qquad (2)$$

in which D = diameter of cylinder, in inches;

p = maximum unbalanced steam pressure on the piston, in pounds per square inch, which should not be taken less than 113;

S_4 = safe bending stress in the material of the wristpin, in pounds per square inch, which, for marine work may be taken as high as 12,500; if the crosshead should be designed for a stationary engine, S_4 had better be kept at a value not greater than 7,000;

P = maximum allowable pressure of the wristpin brass on the wristpin, in pounds per square inch of projected area. For a marine design, P may be made 1,350; for a stationary design, P had better be kept down to 1,200.

6. Let g represent the width of the crosshead shoe, as shown in Fig. 2, and h the length of the crosshead shoe. Then, g is given by the formula

$$g = .11\, D\sqrt{\frac{p}{n}}, \qquad (1)$$

in which n is the ratio of length of connecting-rod to length of crank. The other symbols are the same as in Art. **5.**

The length h is obtained by the formula

$$h = \frac{.7854\, D^2 p}{g P \sqrt{n^2 - 1}}, \qquad (2)$$

in which P is the maximum allowable pressure of the crosshead shoe on the guide, in pounds per square inch. For a marine design, P may be taken at from 40 to 100 on the guide on which the pressure comes when the vessel is going ahead, and as high as 400 on the guide on which the pressure comes when the vessel is backing, if it is important to save weight. For a stationary engine, P had better be kept at a much lower value, not to exceed 30, and as low as 16, if possible. In all cases, the higher the piston speed, the lower should be the value of P.

7. The remaining dimensions of this crosshead are based on the following proportions:

D = diameter of cylinder.
d = diameter of piston rod.
a = $1.75\,d$.
b = $2\,d$.
f = $.05\,D + .5$ inch.
i = $\frac{3}{4}$ inch for D = 15 inches or less; $\frac{7}{8}$ inch for D = from 15 to 20 inches; 1 inch for D = from 20 to 25 inches, and $1\frac{1}{4}$ inches for D above 25 inches.
k must be such that the connecting-rod will clear crosshead and guides.
l = $.05\,D + .25$ inch.
m = .125 inch, constant.
n = $.016\,D$.
o = .5 inch, about.
p = 1.75 inches, about.
r = .125 inch, constant.

8. Fig. 3 shows a modification of the marine crosshead,

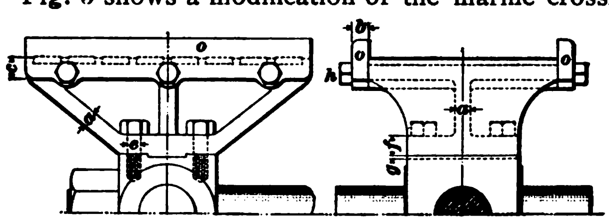


FIG. 3

in which the cast-iron shoe forms a much larger proportion and is provided with guiding strips o, o.

The proportions that apply to this form are:

a = $.03\,D + .5$ inch.
b = $.03\,D + .5$ inch.
c = $.05\,D + .5$ inch.
e = $\frac{7}{8}$ inch up to D = 20 inches; 1 inch up to D = 25 inches; and $1\frac{1}{4}$ inch for D above 25 inches.
f = $.05\,D + .5$ inch.
g = $.02\,D$.
h = $\frac{3}{4}$ inch for D = 20 inches, or less; $\frac{7}{8}$ inch for D = 20 to 25 inches, and 1 inch for D above 25 inches in diameter. Space bolts h not over 7 inches apart.

The other dimensions of this style of crosshead may be obtained from the proportions given for Fig. 2.

The shoes for the crossheads, Figs. 2 and 3, are adjusted by placing liners behind them.

9. Box-Bed Engine Crosshead.—Fig. 4 is an example of a crosshead that is much used on *box-bed engines*. The main part A is made of cast iron and has a boss into which the end of the piston rod is secured by means of a cotter.

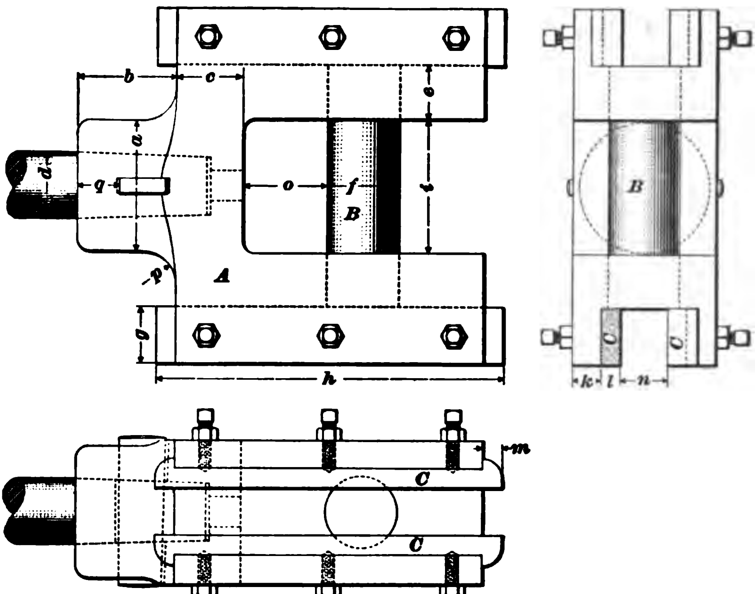


Fig. 4

The wristpin B, which is forced into the crosshead, is usually made of steel, but sometimes of wrought iron. Brass or bronze gibs C, which may be adjusted by means of the set-screws, furnish the surfaces that bear on the guides. The cotters should be designed according to the principles of machine design.

Let f represent the diameter of the wristpin, in inches, as shown in Fig. 4, and i the length of the wristpin journal, in inches. Then, for the diameter and length of a wristpin journal, f and i should be designed according to formulas **1** and **2**, Art. **2**.

The box-bed engine crosshead is used with a large range of speeds. When used with a low-speed engine, all the values may be taken exactly as given by formulas **1** and **2**, Art. **2**, but when used with a high-speed engine, the average value of P in good practice is only about 1,075. An examination of a large number of high-speed stationary engines shows that formula **3**, Art. **2**, applies to this case when the value of K varies between .0518 and .272, with an average value of .0825 in good practice.

10. Let g represent the width of the gib C, in inches, as shown in Fig. 4, which may be taken as $\frac{h}{4}$, and h the length of the gib C, in inches. Then,

$$h^2 = \frac{1.57\,D^2 p}{P\sqrt{n^2 - 1}}$$

The symbols are the same as given in Arts. **3** and **6**.

The following additional values of P are given for use with the formula just stated: With low-speed stationary engines, P varies from about 26 to 58 in good practice, the average being about 36. With high-speed stationary engines, P should be kept down to about 16, if possible. This will often involve making the crosshead excessively large, and in order to avoid this P may be carried up to 25 or even 30, if necessary. With locomotive engines, P varies from 40 to 60. In all cases, the higher the piston speed, the lower should be the value of P.

The other proportions for this type of crosshead are:

D = diameter of cylinder.	$l = \frac{h}{18}$.
d = diameter of piston rod.	$m = l$.
$a = 2\,d$.	n = thickness of guides.
$b = 1.5\,d$.	o = space to clear connecting-rod.
$c = d$.	$p = .5\,d$.
$e = .75\,f$.	$q = .75\,d$.
$k = .075\,D$.	

Setscrews may be $\frac{3}{8}$ inch for cylinders up to 8 inches in diameter; $\frac{7}{16}$ inch for cylinders from 8 to 12 inches in diameter; and $\frac{1}{2}$ inch for all sizes above. Two setscrews should be used for cylinders up to 8 inches in diameter, and three for larger cylinders.

VALVES, VALVE STEMS, AND ECCENTRIC RODS

11. Valve Seats.—The slide valve has been fully considered in *Valve Gears*. Hence, it is necessary here to give only the proportions of the various parts. Two sectional views of a slide valve are shown in Fig. 5. The design of the valve seat will be considered first. The product of the width b and the length l of a steam port is obtained from the formula

$$l b = \frac{A S}{v},$$

in which A = area of piston, in square inches;
S = average piston speed, in feet per minute;
v = average velocity of steam, in feet per minute.

The width of the bridges c, c between the ports is usually made equal to the thickness of the cylinder walls. The width of the exhaust port is from $1\frac{3}{8}$ to $2\frac{1}{2}$ times the width of the steam port. In any given case, the exhaust port must be wide enough so that when the valve is at the end of its travel the width of the portion of exhaust port remaining open is at least equal to the width of the steam port.

12. Let a = width of exhaust port;
b = width of steam port;
c = width of bridge;
k = half of travel of valve;
i = inside lap;
o = outside lap.

Then, in order that the foregoing condition may be fulfilled, a should be equal to or greater than $b + k + i - c$. In Fig. 5,

$$h = a + 2(c - i)$$
$$e = b + i + o$$
$$L = h + 2e = a + 2c + 2b + 2o$$
$$B = l + 2t$$

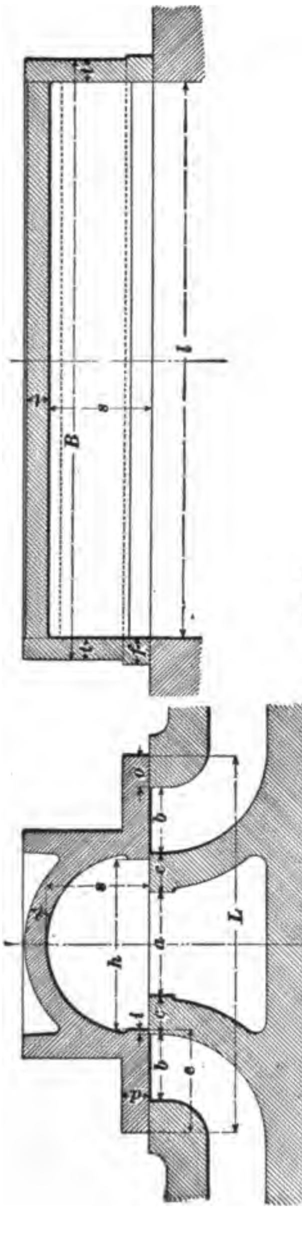


FIG. 5

When the valve at the end of its travel just opens the steam port fully, $k = b + o$.

The height s of the hollow underneath the valve must be sufficient to allow a free exhaust. For low piston speeds, s may be made equal to or slightly greater than the width b of the steam port; that is, $s = b$. More often, $s = \frac{3}{4} a$ to a, where a is the width of the exhaust port. Also, d may be $1.2\,t$, and f may be $1.1\,t$.

The thickness t of the metal of the valve equals

$t = .03\ D + .25$ inch (cast iron),

where D is the cylinder diameter, in inches.

The lead, lap, valve travel, etc. are readily determined from the valve diagram (see *Valve Gears*).

These quantities, in addition to the proportions just given, furnish sufficient data to design a plain slide valve, and may also be readily applied to the design of more complicated forms of slide valves, such as the double-ported valve, piston valve, etc.

13. The Valve Stem.—The valve stem must be designed to move the valve under the most unfavorable conditions that may occur in practice; hence, it may be assumed that the valve is unbalanced, for, even if balanced, the joint may leak. Furthermore, the valve may run dry on the seat, thus increasing the friction.

Let B = breadth of valve, in inches, as shown in Fig. 5;
d = diameter of valve stem, in inches;
E = modulus of elasticity of valve-stem material;
f' = coefficient of friction;
L = length of valve, in inches;
l = length of valve stem, in inches;
p = pressure on back of valve, in pounds per square inch, which may be taken as absolute boiler pressure;
f = factor of safety.

Then the load that the valve stem must move is equal to $f' p B L$ pounds.

Under the most favorable circumstances, f' should not exceed .2, but for designing purposes, f' should be made .25, as the lubrication may fail. Then, the load on the valve stem is $.25\, p B L$ pounds.

As the valve stem is alternately under tension and compression, it should be designed as a column under compression and should be considered as pin-ended at both ends.

The diameter of the valve stem is therefore given by the formula

$$d = .85\sqrt[4]{\frac{p B L l^2 f}{E}} \qquad (1)$$

As the load on the valve stem is generally much less than that just assumed, except under extraordinary conditions, f may be given a value of about 6. Then, taking E as 30,000,000 for steel and 25,000,000 for wrought iron, formula **1** becomes

$$d = K\sqrt[4]{p B L l^2}, \qquad (2)$$

in which K is equal to .018 for steel and .0194 for wrought iron.

EXAMPLE.—Find the diameter of a steel valve stem 17 inches long for a locomotive valve 18.5 in. × 10.25 in. on the face, the boiler pressure being 150 pounds, gauge.

SOLUTION.—Applying formula **2**,

$$d = .018\sqrt[4]{164.7 \times 18.5 \times 10.25 \times 17^2} = 1 \text{ in., nearly.} \quad \textbf{Ans.}$$

14. Valve-Stem Fastening.—Valve stems are fastened to valves in different ways. Fig. 6 shows a steel yoke suitable for a simple slide valve like that shown in Fig. 5.

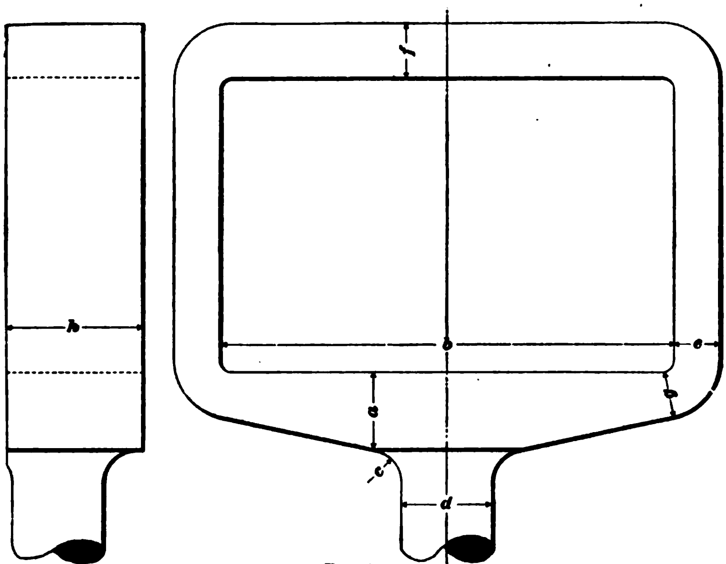


FIG. 6

The following proportions will show the proper dimensions. Where the meaning of a symbol is not stated, it is the same as given in Art. **13.**

$a = \frac{4f}{3}$.

b = dimension to fit valve.

$c = .375\,d$.

d = diameter of valve stem, in inches.

$e = \frac{p\,B\,L}{24{,}000\,h}$.

$f = .0025\sqrt{\frac{p\,B\,b\,L}{h}}$.

$g = .5\,d$.

h = dimension to fit valve.

15. Two other methods of fastening the valve to the stem are shown in Fig. 7. In (*a*), the valve stem is forged with two collars, and in (*b*) the end of the stem is threaded for two sets of nuts. Either arrangement allows the valve to adjust itself somewhat and thus prevents the stuffingbox from

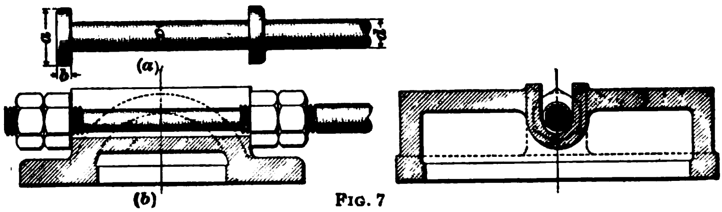


FIG. 7

wearing. The arrangement in (*b*) also furnishes a means for setting the valve, so as to get the proper location over the ports.

The proportions for the valve-stem fastenings shown in Fig. 7 are:

$c = d.$

$a = 2d.$

$b = \dfrac{pBL}{62{,}800\,c}.$

d = diameter of valve stem.

The values of p, B, and L are as stated in Art. **13.**

In the arrangement shown in Fig. 7 (*b*), the thickness of each nut should be at least as great as *b*. The outer end of the valve stem may terminate in some form of crosshead running in guides or it may be jointed to a rocker-arm.

16. Eccentric Rods.—Eccentric rods may be rectangular or circular in cross-section. Quite often the rod is tapered, being largest where it joins the eccentric strap. It is common practice to make the area of the smallest section of the rod equal to about .8 the area of cross-section of the valve stem, the latter being calculated by either formula **1** or formula **2,** Art. **13.** The area at the large end may then be made about one-third larger than the area at the small end.

Fig. 8 shows an eccentric rod with right and left threaded ends, one of which is attached to the eccentric strap, and the other to a brass bearing for connecting to the valve-rod pin or rocker-arm pin. The threaded ends of the rod furnish means for adjusting the valve, and the locknuts *N* prevent the rod from turning after the valve is properly set.

The brass has a loose piece *O*, which can be adjusted by the cotter, thus furnishing means of taking up wear.

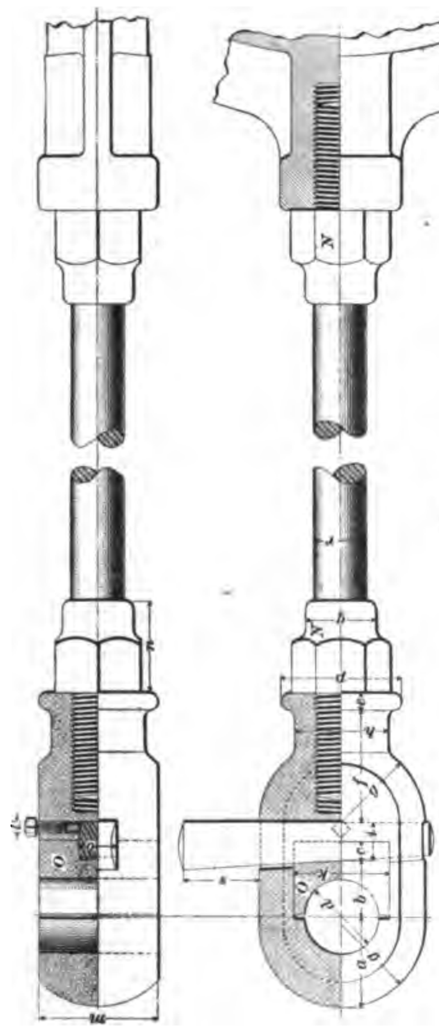


Fig. 8

The following proportions will give the dimensions for this rod and its brass:

D = diameter of valve stem.
$d = 1.77\ D$.
$a = 1.2\ d$.
$b = .75\ d$.
$c = .25\ d$.
$e = .3\ r$.
$f = 1.75\ d$.
$g = 1.1\ d$.
$h = 1.75\ r$.
$i = .25\ d + .25\ D + .1875$ inch, but never less than $.25 d + .4375$ inch.
$k = 1.3\ d$.
$l = .25\ D$, but never less than .25 inch.
$m = 1.6\ d$.
$n = 1.75\ r$.
$o = .5\ d$.
$p = 2.2\ r$.
$q = 1.3\ r$.
r to be designed as a long column.
$s = d$.

Taper of cotter = $\frac{3}{4}$ inch per foot.

17. Fig. 9 shows an eccentric rod with a modification of the marine connecting-rod end for the valve-stem pin bearing.

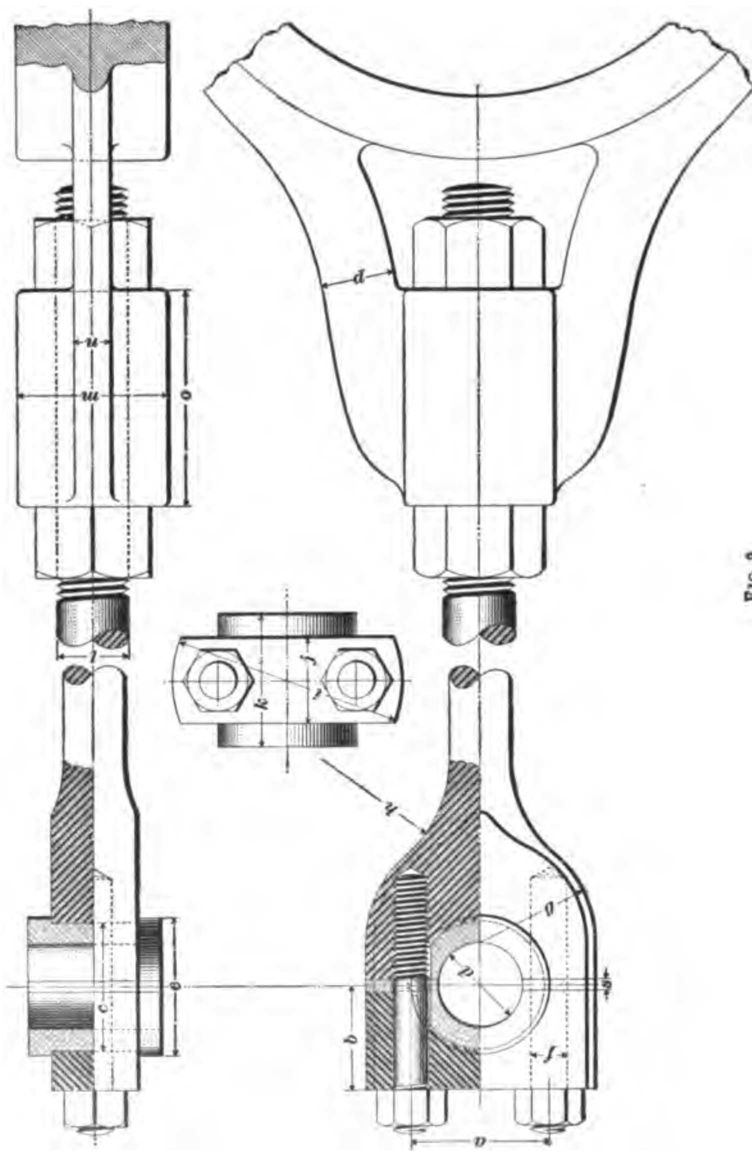


FIG. 9

The rod passes through a boss on the eccentric strap, and is fastened by the two nuts. This construction permits the valve to be adjusted when necessary. The bearing for the valve-stem pin is composed of brasses held in place by a wrought-iron cap and stud bolts. Liners are placed between the end of the rod and the cap, and the brasses may be adjusted by filing.

The proportions for this rod and for the boss that fastens it to the eccentric strap are as follows:

D = diameter of valve stem.
$d = 1.77\,D$.
$a = d + f + .375$ inch.
$b = 1.25\,d$.
$c = 1.5\,d$.
$e = 1.5\,d + .1875$ inch.
f = area at root of thread $= .38\,D^2$.
$g = .5\,i$.
$h = 1.5\,d$.
$i = 3.5\,f + d + .375$ inch.
$j = d$.
$k = 1.6\,d$.
l = diameter of eccentric rod at eccentric end.
$m = 2.1\,l$.
$n = .75\,D$.
$o = 3\,l$.
$p = 1.5\,D$.
$s = .125\,d$.

The diameter of the rod is determined by treating it as a long column.

ECCENTRIC SHEAVES AND STRAPS

18. An eccentric especially adapted for vertical engines is shown in Fig. 10; it has a cast-iron sheave and a steel strap. The sheave is made in two parts, so that it can be put on or removed from the shaft without disturbing flywheels or bearings. This is sometimes necessary, owing to the construction of the shaft, which will not permit an eccentric to be slipped into place over the end. The end of the eccentric rod is forged T-shaped and is fastened to the strap by tap bolts.

The two halves of the strap are held apart by liners, which permit of adjustment for wear. Split pins are put through

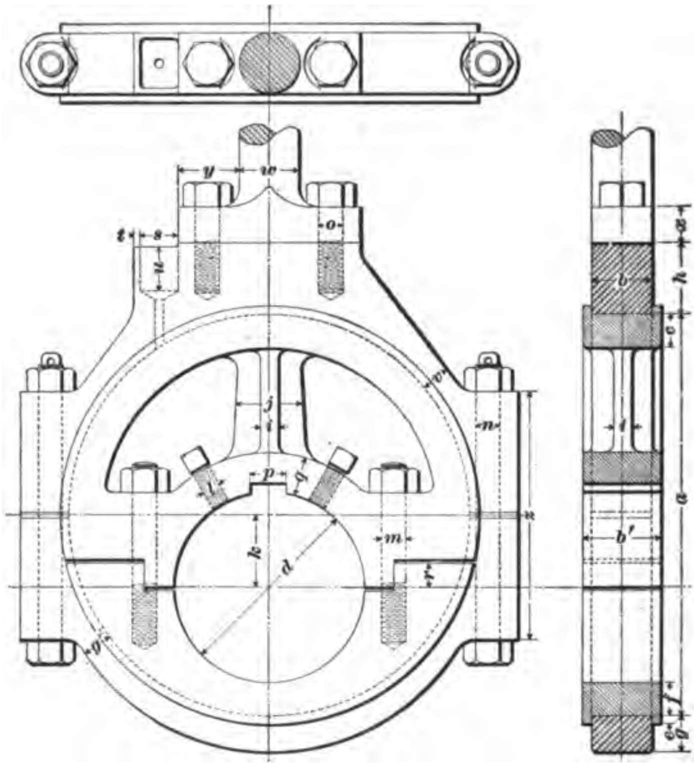


FIG. 10

the holes in the ends of the bolts n, to keep the nuts from turning off.

The following proportions give the necessary dimensions for this eccentric:

D = diameter of valve stem.

d = diameter of shaft.

$a = d + 2k + 2f$, never less.

$b = 2.5n$ at least, but never less than w.

$b' = b + .5c$.

$c = .6D + .375$ inch.

$e = .25c$.

f = .7 D + .5 inch, unless more is required to allow nuts to be placed on stud m.

g = .6 D + .375 inch.

g' = .5 D + .25 inch.

h = 3 o.

i = .5 c.

j = 2 c.

k = eccentricity.

l = .75 of diameter of bolt m.

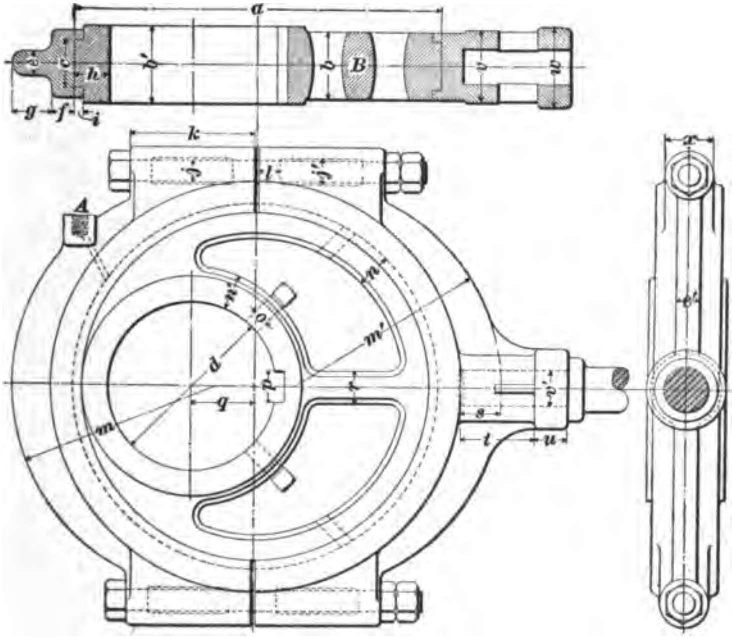


FIG. 11

Area of bolt m at root of thread = .38 D^2; use nearest standard size of bolt.

n = diameter of bolt m.

o = diameter of bolt m.

p = .7 D + .5 inch.

q = .7 D + .5 inch.

r = .125 d.

s = D.

t = .25 inch, constant.
u = 1.25 D.
v = .5 D + .25 inch.
w = diameter of eccentric rod.
x = .6 w.
y = 2.5 o.

z is to be found by laying out. The bolt n should clear the eccentric sheave by $\frac{1}{4}$ inch on all sizes up to D = $1\frac{1}{2}$ inches; $\frac{3}{8}$ inch for sizes of D from $1\frac{1}{2}$ to 2 inches; and $\frac{1}{2}$ inch for all sizes above D = 2 inches.

19. In Fig. 11, both the eccentric sheave and the strap are made of cast iron. The eccentric sheave is cast solid, and must therefore be slipped over the end of the shaft. The eccentric rod is held in a boss on the strap by means of a cotter.

It will be seen that in this case the strap is grooved for the sheave, while in Fig. 10 the groove is in the sheave. The construction that places the groove in the strap has the advantage of retaining oil better.

For eccentrics used with valve stems $\frac{1}{2}$ inch or less in diameter, the holes for bolts j are not to be cored.

A shows the boss for the oil cup, and B the cross-section of rib r.

The proportions are:

D = diameter of valve stem.
d = diameter of shaft.
$a = d + 2q + 2h$.
b = 2 D + .125 inch.
b' = 2.25 D + .125 inch.
c = 1.5 D.
e = .75 D.
e' = .75 D.
f = .7 D.
g = 1.25 D.
h = D + .125 inch.
i = .25 D + .0625 inch.

j = area of bolt at root of thread = .38 D^2; use the nearest standard size bolt.
j' = j + .1875 inch.
k = 4 D.
$l = j$.
$m = \dfrac{d + 2q + 2h + 2f}{2}$.
$m' = m$.
n = D + .125 inch.
n' = D + .125 inch.
o = .75 j.

$p = D$. $u = D$.
q = eccentricity. $v = 2.25\ D$.
$r = D$. $v' = 1.125\ D$.
$s = 1.25\ D$. $w = 2.5\ D$.
$t = 2.25\ D + 1.25$ inch. $x = 2.25\ j$.

STUFFINGBOXES

20. A **stuffingbox** of the ordinary form is shown in Fig. 12. The gland may be made of brass, of cast iron lined with brass, or simply of cast iron. The brass lining, however, injures the rod less than the harder iron. The gland is usually held in place by two stud bolts, but for

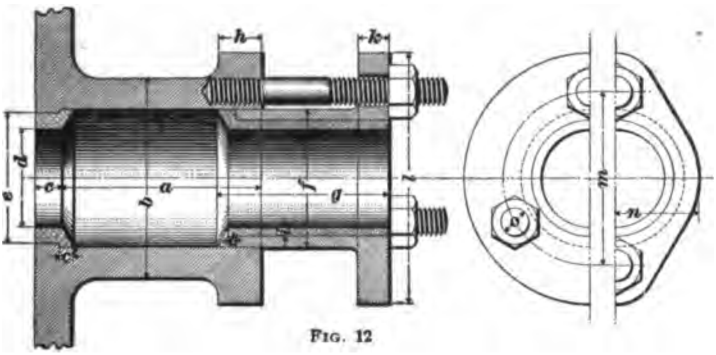


FIG. 12

large rods the gland is sometimes made circular instead of oval, and fastened by three or more studs.

The proportions for the gland shown in Fig. 12 are:

d = diameter of rod. $g = 1.5\ d + 1$ inch.
$a = 1.6\ d + 1.5$ inches. $h = .3\ d + .5$ inch.
$b = 1.75\ d + 1.125$ inches. $i = .04\ d + .1875$ inch.
$c = .1\ d + .75$ inch. $k = .25\ d + .25$ inch.
$c' = .5\ c$. $l = 2.25\ d + 1.75$ inches.
$e = 1.25\ d + .375$ inch. $m = 1.6\ d + 1.25$ inches.
$f = 1.25\ d + .625$ inch. $n = .75\ d + .375$ inch.

$o = .25\ d + .25$ inch for two bolts.
$= .2\ d + .25$ inch for three bolts.
$= .05\ d + 1.0625$ inches for four bolts.

$t = i$.

Two bolts should be used for glands on rods up to 3.5 inches in diameter. Above that size the gland should be made round, while three bolts should be used for rods up to 5.5 inches in diameter, and four bolts for all larger sizes.

21. For very high steam pressures, various styles of metallic piston-rod packing are used, one form of which is shown in Fig. 13. The construction of the stuffingbox and gland for this packing is very similar to the form shown in Fig. 12, but the packing is made up of rings of brass or similar antifriction metal. These rings are made in two or

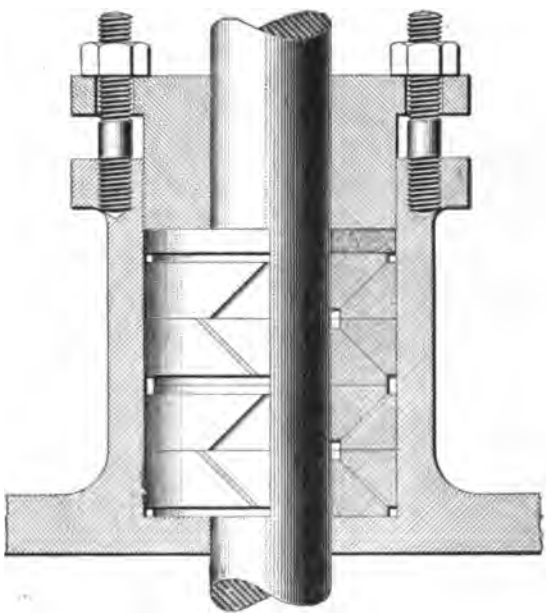


FIG. 13

more segments, depending on the size of the rod. By reason of the conical shape of the rings, the pressure of the gland forces the inner ones against the rod, while the outer ones are pressed against the sides of the stuffingbox. A rubber or fibrous ring placed between the gland and the first ring serves as a cushion to make the packing slightly elastic.

22. A stuffingbox of the form shown in Fig. 14 is generally used for small work, such as the spindles of valves, etc. The outside of the stuffingbox is threaded to receive a hexagonal nut, which fits over the gland. As the nut is screwed down, the gland is pressed downwards and compresses the packing.

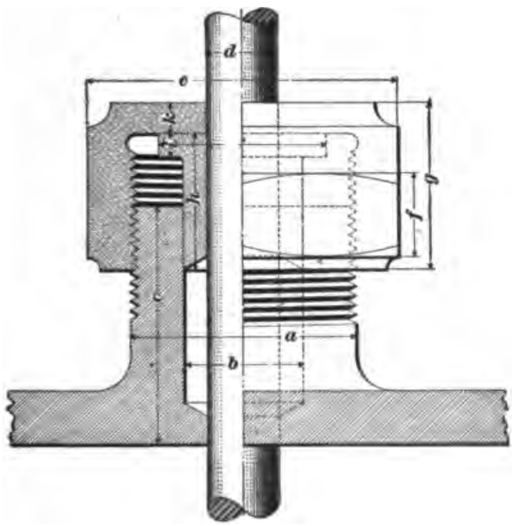


FIG. 14

The proportions used are:

d = diameter of rod.	$f = d + .125$ inch.
$a = 2.5\,d + .5$ inch.	$g = 2\,d + .25$ inch.
$b = 1.5\,d + .125$ inch.	$h = 1.5\,d + .25$ inch.
$c = 3\,d + .25$ inch.	$i = .25\,d + .0625$ inch.
$e = 3.5\,d + .625$ inch.	$k = .5\,d$.

This design may be used for rods up to $1\frac{1}{4}$ inches in diameter.

The number of threads per inch should be made the same as for a bolt having a diameter equal to the diameter of the rod.

ENGINE FLYWHEELS

23. Flywheels are subjected to a variety of complicated stresses, and it is therefore impossible to base their design on theory alone. Empirical rules representing successful practice are also an unsafe guide outside of the range of practice from which they are deduced, and, further, this range is generally unknown to the designer. The most satisfactory method is to work out the required weight of the flywheel and some of the principal dimensions rationally, and then calculate the detail dimensions by empirical rules deduced from practice. After this, the most important stresses may be calculated rationally, and, finally, if the last computation shows that the factor of safety is too small, such modifications should be made as will increase the factor to the necessary amount.

Small flywheels are generally one solid casting, the limit to the size of this construction being about 10 feet in diameter; but, even in this small size, the hub is often split so as to relieve cooling stresses in the casting. The two halves of the hub must then be given some mechanical connection. Wheels from 10 to 15 feet in diameter are generally cast in halves, while still larger wheels are usually cast in several pieces, the hub being cast separate and in halves, the arms separate, and the rim in at least two segments. These pieces are usually connected by bolts and links.

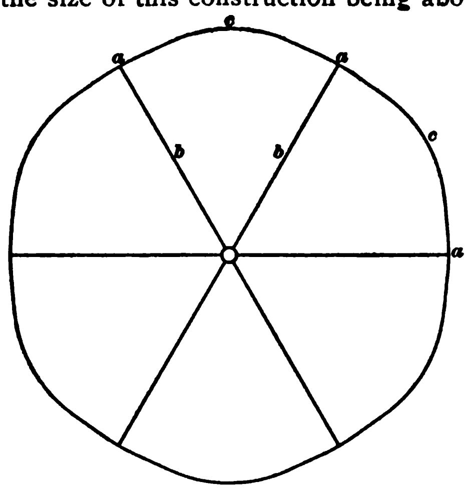


FIG. 15

24. Distortion of Flywheel Rim.—A thin ring rotating about its own center would be subject to the action of centrifugal force, which would act outwards uniformly all over the ring. Under this section, the ring, if composed of

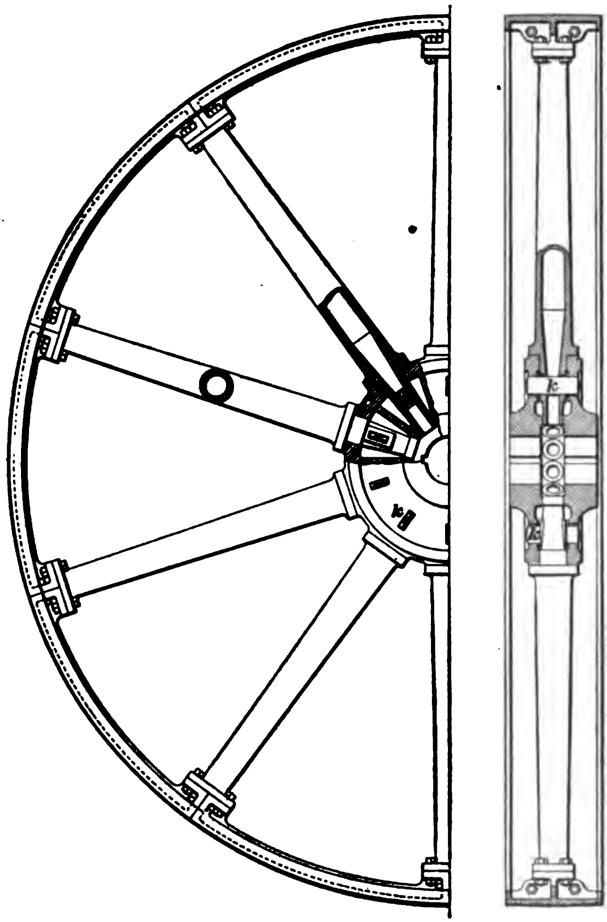


FIG. 16

elastic material, would expand. Now, if, at certain points, the ring should be held to its original diameter, as by the arms of a flywheel, it would take the distorted form shown somewhat exaggerated in Fig. 15. The rim is restrained at the points a, a by the arms b, b, and is bent outwards at c, c by centrifugal force. In the actual flywheel, the action is modified by the fact that the arms themselves stretch outwards under their own centrifugal force and the pull exerted on them by the rim, but the general effect remains as shown.

It is apparent from these remarks that a rotating flywheel with a small number of heavy arms will suffer greater distortion than one with a large number of light arms having the same total strength. In other words, the farther apart the arms, the greater the distortion of the rim between the arms.

25. A common form of construction for large flywheels is shown in Fig. 16. Particular attention is here called to the manner of attaching the segments of the rim to each other and to the arms. This method is superior to that in

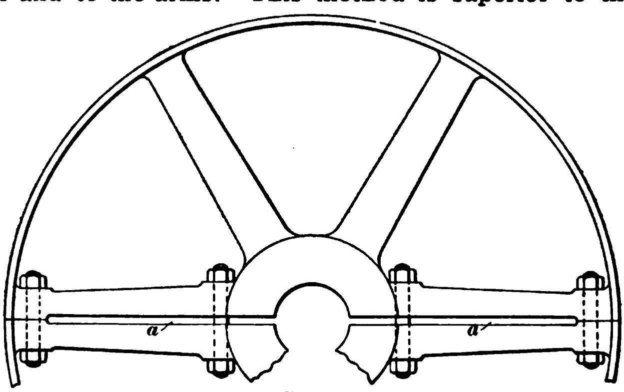


FIG. 17

which the segments of the rim are bolted together half way between the arms. The attachments of the rim to the arms as shown in the figure do not weaken the rim, but, on the contrary, tend to stiffen it and make the joint at the arms as strong as the solid rim. Furthermore, it is apparent from Fig. 15 that the distortion of a rotating wheel tends to open a joint half way between the arms, while it tends to close a radial joint directly over the arm.

If the joint in the rim cannot be placed directly over the arm, the best place for it is at a distance from the middle of the arm equal to .211 of the distance between middle lines of adjacent arms, measured along the middle of the rim. This value was determined by regarding the section of the rim from one arm to the next as a beam fixed at the ends and uniformly loaded, and finding, by means of higher mathematics, the point where the bending moment is zero.

26. Wheels in halves should have double arms along the line of separation, as shown at *a*, *a*, Fig. 17, especially if the rims are thin. The spokes, or arms, of flywheels are generally made of elliptical cross-section. In good practice, the major axis of the ellipse is placed in the plane of rotation, and is usually made from two to three times the length of the minor axis. For a high-speed wheel, an arm in which the minor axis is short has the advantage over one in which it is long, as the thinner and more wedge-shaped sections have much less air resistance. This is an important feature in a wheel traveling at a rim speed of about a mile a minute. The longer major axis, which is used with the shorter minor axis, also gives the arms greater resistance to the bending actions to which they are subjected through belt pull or variations in the speed of the wheel. The arms generally taper from the hub to the rim. The maximum taper found in good practice is such that the cross-sectional area at the rim is two-thirds of that at the hub.

27. Tension in Flywheel Rim.—In the design of a flywheel, the first value to be determined is the linear velocity at which the rim is to be run.

Let v = linear velocity of center of gravity of rim section, in feet per second; this center of gravity may be taken with sufficient accuracy as being at the mean rim radius from the center of the shaft;

S = stress per unit area of cross-section of rim;

H = weight of material in rim, in pounds per cubic foot.

The stress in the flywheel rim depends on the weight per cubic foot of the material in the rim and on the velocity of the mean rim section. From the principles of mechanics and strength of materials, the unit stress in the flywheel rim due to centrifugal force is expressed by the formula

$$S = \frac{H v^2}{4{,}631} \qquad (1)$$

In the case of cast iron, $H = 450$, and formula **1** reduces to

$$S = .09717\, v^2 \qquad (2)$$

For convenience in making calculations, formula **2** is frequently expressed by the approximate formula

$$S = .1\, v^2 \qquad (3)$$

Multiplying both sides of formula **1** by the area of cross-section of the rim in square inches, it becomes

$$T = SA = \frac{H A v^2}{4{,}631}, \qquad (4)$$

in which T = total tension in a cross-section of the rim, in pounds;
A = area of cross-section of rim, in square inches.

EXAMPLE.—What is the stress in the rim of a cast-iron flywheel when it has a velocity of 80 feet per second?

SOLUTION.—Apply formula **2**. Substituting $v = 80$, then
$S = .09717 \times 80 \times 80 = 621.9$ lb. per sq. in. Ans.

28. Cheapness of construction would always lead to building a wheel with a light rim of large diameter, rather than one with a heavier rim of smaller diameter. That is, high rim speed is desirable on the ground of economy or low first cost; but, according to the formulas of Art. **27**, the stress in the material increases as the square of this speed, and a limit is soon reached, beyond which it is not safe to go.

In good practice, the average velocity of flywheel rims, at least with wheels of moderate size cast in one piece, is about 70 feet per second, which, for cast iron, makes $S = 475$, nearly. Larger wheels cast in several pieces and bolted together are often run at a speed of about 88 feet per second, making $S = 750$, nearly. These limits of speed cannot be safely exceeded with cast-iron wheels of ordinary construction. If higher speeds are necessary, material of greater tensile strength in proportion to its weight must be employed for the rim, and special constructions must be used in the arms and hub.

29. Size of Flywheel Rim.—If the value of v is decided on, the weight of the wheel can be calculated. It is customary to base the calculations on the supposition that the entire weight is in the rim. The weight of the arms and hub is then additional. This results in making the wheel regulate the speed somewhat closer than was calculated, which is an advantage. The required weight of the rim for any given case is found as shown in *Mechanics of the Steam Engine.*

Let D = mean diameter of rim in feet $= \frac{60\,v}{\pi\,N}$;
v = velocity of rim, in feet per second;
N = number of revolutions per minute;
π = 3.1416;
W = weight of rim, in pounds;
h = weight of rim material per cubic inch, which is .261 for cast iron;
A = area of cross-section of rim, in square inches.

Then, the volume of the rim in cubic feet is $\frac{\pi\,D\,A}{144}$, and the weight of the rim is 1,728 h times this quantity, which gives the formula:

$$W = 12\,\pi\,D\,A\,h \qquad (1)$$

or,

$$A = \frac{W}{12\,\pi\,D\,h} \qquad (2)$$

EXAMPLE.—What is the area of cross-section of rim of a flywheel having a mean diameter of 12 feet, if the rim is required to weigh 5,288 pounds?

SOLUTION.—Applying formula **2**, the cross-section of the rim is

$$A = \frac{5{,}288}{12 \times 3.1416 \times 12 \times .261} = 44.8 \text{ sq. in. Ans.}$$

30. If the wheel is not to carry a belt, the rim will probably be approximately rectangular in cross-section, the depth being usually from 1.1 to 1.4 times the breadth. The usual form of the rim for such a wheel is shown in Fig. 18. This illustration also shows the general construction for a small wheel to be run at low velocity; the whole wheel is one casting. In this construction, cooling stresses from casting are sure to occur, and are likely to exceed in magnitude the stresses that can be calculated. Also, the interior of a heavy rim is liable to have porous spots and other defects that cannot be detected. The only way to provide against accident due to these defects in a wheel of this kind is to keep down its rim speed, which should therefore never

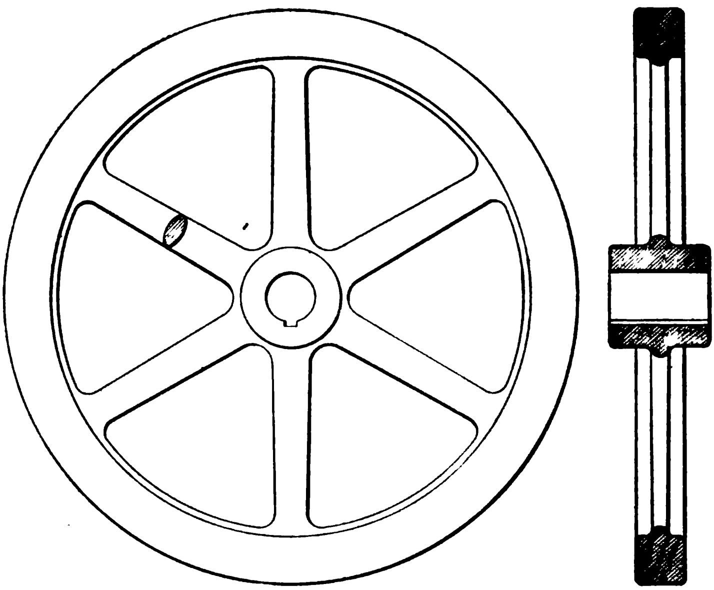


FIG. 18

exceed a limit of 70 feet per second, as stated in Art. **28.** Further calculation of stresses in the rim of a wheel of this construction is practically useless.

31. If a belt is to run on the flywheel, the width of the wheel must be designed to suit the belt.

Let D' = outer diameter of flywheel, in feet;

D = mean diameter of rim, in feet;

d = thickness of belt, in inches; for convenience, d will be taken as $\frac{1}{4}$ inch for a single belt, and $\frac{1}{2}$ inch for a double belt, although the actual thicknesses may vary slightly from these values;

f = coefficient of friction between belt and wheel, which may be taken as .25 in a design of this kind;
N = number of revolutions of the engine per minute;
P = maximum allowable tension on belt, in pounds per square inch of cross-section, which is from 250 to 400, with 300 as an average value;
T_1 = tension on tight side of belt, in pounds;
T_2 = tension on loose side of belt, in pounds;
V = velocity of outside of rim, in feet per minute;
t = thickness of flywheel rim, in inches, so that $D' = D + t$;
w = width of flywheel rim, in inches;
w_1 = width of belt, in inches;
h_1 = weight of belt in pounds per cubic inch, which may be taken as .035;
a = arc of contact between belt and wheel rim, in degrees;
H. P. = horsepower of engine.

It is first necessary to assume a value of t. This is done according to the best judgment of the designer, and at once gives the value of D', as D has already been found.

The difference in the tension on the two sides of the belt is $T_1 - T_2$, the belt pull, in pounds. As the whole of the horsepower of the engine is transmitted through this belt, the pull is also equal to the number of foot-pounds transmitted, 33,000 $\times$ H. P., divided by the number of feet passed through, $\pi D' N$. Hence, these two expressions for belt pull give the formula

$$T_1 - T_2 = \frac{33{,}000 \times \text{H. P.}}{\pi D' N} \qquad (1)$$

In order to determine the cross-sectional area of the belt required to transmit a given horsepower, it is necessary to ascertain the maximum tension T_1. As formula **1** gives only the difference $T_1 - T_2$, it is necessary to use another formula in connection with formula **1**. This formula, which is derived by means of higher mathematics, is

$$\log \frac{T_1}{T_2} = 2.729 f(1 - z) \frac{a}{360}, \qquad (2)$$

in which z is a factor depending on the centrifugal force of the belt. The other factors are as already given.

By means of higher mathematics, the value of z is found to be expressed by the formula

$$z = \frac{h_1 V^2}{9{,}660 P} \qquad \textbf{(3)}$$

Then, for a velocity of 70 feet per second, when $P = 300$, the value of z is .213, and for 88 feet per second, $z = .337$, nearly.

By using formulas **1** and **2**, the value of T_1 may be found. Then, the width of the belt is found by the formula

$$w_1 = \frac{T_1}{d P} \qquad \textbf{(4)}$$

The rim of the flywheel is usually made from $\frac{1}{2}$ to 1 inch wider than the belt. If the product of w as thus found and that of t as previously assumed is not sufficiently near the area of the cross-section of the rim as found by Art. **29**, a new value of t must be assumed, and the process repeated until the result is satisfactory.

EXAMPLE.—A flywheel having an average rim diameter of 12 feet and a rim section of 44.8 square inches is to be used on an engine of 200 horsepower at full load. The engine runs at 120 revolutions per minute, and the angle of contact is 180°. Find the dimensions of a flywheel rim of rectangular section to carry a double belt.

SOLUTION.—Assume that t = 4 in., or $\frac{1}{3}$ ft. Then, $D' = 12 + \frac{1}{3}$ = 12.33 ft., and the velocity V becomes

$$\pi D' N = 3.1416 \times 12\tfrac{1}{3} \times 120 = 4{,}650 \text{ ft. per min.}$$

Then, by using formula **3**, making $h_1 = .035$, $V^2 = 4{,}650 \times 4{,}650$, and $P = 300$,

$$z = \frac{.035 \times 4{,}650 \times 4{,}650}{9{,}660 \times 300} = .261$$

Applying formula **2**, making $f = .25$, $z = .261$, and $a = 180$,

$$\log \frac{T_1}{T_2} = 2.729 \times .25\ (1 - .261) \frac{180}{360} = .25209,$$

and the number whose log is .25209 is 1.787.

Hence, $$\frac{T_1}{T_2} = 1.787 \text{ and } T_2 = \frac{T_1}{1.787}$$

Applying formula **1**, making H. P. = 200, and $\pi D' N = 4{,}650$,

$$T_1 - \frac{T_1}{1.787} = \frac{33{,}000 \times 200}{4{,}650}$$

Hence, $$T_1 = 3{,}223 \text{ lb.}$$

To find the width, apply formula **4**, making $T_1 = 3{,}223$, $d = .5$, and $P = 300$. Thus,

$$w_1 = \frac{3{,}223}{.5 \times 300} = 21\tfrac{1}{2} \text{ in., nearly.}$$

Hence, a 22-in. double belt should be used.

Then, $w = 22.5$ in., at least, and the area of the cross-section $= 4 \times 22\tfrac{1}{2} = 90$ sq. in., instead of the 44.8 sq. in. necessary. It is seen, therefore, that a 4-in. rim is about twice as thick as it should be. Therefore, try $t = 2$ in. Going through the calculations the same as before, w_1 now becomes 21.6. As before, a 22-in. belt will be used.

To get 44.8 sq. in.,

$$w = \frac{44.8}{2} = 22.4 \text{ in. Ans.}$$

This may be increased to 22.5 in., and used with a 22-in. belt.

32. Flywheel Arms.—The dimensions of the arms, or spokes, of the flywheel may now be calculated. The turning of the shaft by the engine is resisted by the belt pull and the inertia of the flywheel rim, and this produces a bending stress in the arms. Each arm may then be considered as a cantilever loaded at one end. Then, by finding expressions for the bending moment and for the resisting moment of the section of the arm nearest the hub, and placing them equal to each other, a formula for the size of arm is derived.

Let S_4 = safe bending stress in outer fiber of arm, at hub, in pounds per square inch; S_4 should not exceed 1,000 to 1,400 for cast-iron arms;

N = number of revolutions of flywheel per minute;

n = number of arms in wheel;

T = mean twisting moment, in inch-pounds, transmitted by shaft in ordinary driving, which tends to bend the arms about the hub;

I = moment of inertia of cross-section of arms, dimensions in inches;

c = distance from neutral axis to outside edge, in inches.

The twisting moment of the flywheel is equal to the total bending moment on all the arms and also to the product of the difference between the belt tensions, or the equivalent from the power transmitted, and the radius of the flywheel in inches. It would be correct to use the length of the arms, but the radius is more convenient and its use has the effect of increasing the factor of safety. Hence, multiplying formula **1**, Art. **31**, by the outside radius in inches gives the formula for T as

$$T = \frac{198{,}000 \text{ H. P.}}{\pi N} \quad (1)$$

In a flywheel having a heavy rim, it is reasonable to assume that the turning moment is divided equally among the arms. Hence, the bending moment on each arm is $T \div n$, regarding the arm as a beam fixed at one end and loaded at the other. Placing this equal to the resisting moment of the arm,

$$\left.\begin{aligned} \frac{T}{n} &= S_4 \frac{I}{c} \\ \text{or,} \quad \frac{I}{c} &= \frac{T}{n S_4} \end{aligned}\right\} \quad (2)$$

Let A = outside depth, that is, the dimension in the direction of motion, of arm at hub, in inches;
a = inside depth of arm, if hollow, at hub, in inches;
B = outside breadth, that is, the dimension perpendicular to the plane of rotation, of arm, at hub, in inches;
b = inside breadth of arm, if hollow, at hub, in inches.

For some of the simple forms, the values of $\frac{I}{c}$ are:

For a solid rectangle, $\frac{I}{c} = \frac{A^2 B}{6}$.

For a hollow rectangle, $\frac{I}{c} = \frac{1}{6} \frac{(B A^3 - b a^3)}{A}$.

For a solid ellipse, $\frac{I}{c} = \frac{\pi}{32} B A^2$.

For a hollow ellipse, $\frac{I}{c} = \frac{\pi}{32} \frac{(A^3 B - a^3 b)}{A}$.

A may be taken as from $2B$ to $3B$, while suitable relations must be assumed between the interior and exterior dimensions of the cross-section, if the arm is hollow. The dimensions of the cross-section may then be determined by use of formulas **1** and **2**.

EXAMPLE.—A flywheel makes 120 revolutions per minute in transmitting 200 horsepower. If the power is transmitted through a flywheel with eight solid elliptical arms, with the major axis $2\frac{1}{2}$ times the minor axis, what are the dimensions of the arms at the hub?

SOLUTION.—Applying formula **1**, making H. P. = 200, $N = 120$, and $\pi = 3.1416$,

$$T = \frac{198{,}000 \times 200}{3.1416 \times 120} = 105{,}042$$

When $A = 2\frac{1}{2}B$, the value of $\frac{I}{c}$ is

$$\frac{\pi}{32}BA^2 = \frac{\pi \times 6.25\,B^3}{32} = .6136\,B^3$$

Then, applying formula **2**, making $\frac{I}{c} = .6136B^3$, $S_t = 1{,}000$, $n = 8$, and $T = 105{,}042$,

$$.6136\,B^3 = \frac{105{,}042}{8 \times 1{,}000} = 13.13$$

Hence, $$B = \sqrt[3]{\frac{13.13}{.6136}} = 2.776 \text{ in.}$$

Then, $A = 2\frac{1}{2}B = 2\frac{1}{2} \times 2.776 = 6.94$, say 7, in.

Hence, at the hub, the arms would probably be $2\frac{7}{8}$, or 3 in. $\times$ 7 in. Ans.

33. Check Calculations for Built-Up Flywheels. In the foregoing calculations, the effect of the mutual pull of rim and spokes on each other has been entirely neglected. This pull, however, produces serious stresses, and to compensate for the neglect of these, the calculated stresses have been made very low. In the case of flywheels, this method of designing has sometimes been unsatisfactory, and the bursting of large flywheels has not been uncommon.

Therefore, in any built-up flywheel, except where the design is practically a repetition of one that has already proved safe in actual service, the following stresses should be calculated: (1) The tensile stress in the inner fibers of the rim directly over the arms; (2) the tensile stress in the outer fibers of the rim, half way between the arms; (3) the tensile stress in the outer fibers of the arms at the hub; and (4) the tensile stress in the arm at the rim.

Though the calculation of these stresses is somewhat laborious, it should be done for any built-up wheel except those just stated. In deriving the following formulas for the principal stresses, the arms were assumed to be of uniform cross-section throughout. In practice, the arms are frequently tapered; but the formulas are probably a close enough approximation to the real values of the stresses with tapered arms, provided the taper does not exceed that stated in Art. **26.** However, this is another reason why the stresses as calculated should be kept very low, as compared with those allowed in other portions of the engine.

34. The following formulas are based on formulas for the stresses in a built-up flywheel as derived by Professor Gaetano Lanza. They are derived by means of higher mathematics from a careful analysis of all the stresses in the flywheel due to the centrifugal force of the rim and the restraining effect of the arms.

Let A = area of cross-section of rim, in square feet;

A_1 = area of cross-section of arm at hub, in square feet;

A_2 = area of cross-section of arm at rim, in square feet;

H = weight of material, in pounds per cubic foot;

g = 32.16;

I = moment of inertia of cross-section of rim or arm about its neutral axis, the dimensions of the section being taken in feet;

c = distance from neutral axis to outermost fiber of section, in feet;

n = number of arms in wheel;

R = distance from center of hub to middle of cross-section of rim, in feet;

r_1 = distance from center of hub to outer end of arm, in feet;

r_2 = radius of hub, in feet;

T = mean twisting moment transmitted by shaft, in foot-pounds;

v = linear velocity of middle of cross-section of rim, in feet per second;

y_1 = distance from neutral axis of rim to inside of rim, in feet;

y_2 = distance from neutral axis of rim to outside of rim, in feet;

a = half the angle between middle lines of two adjacent arms;

K = a constant.

NOTE.—If the cross-section of the rim is symmetrical about a line through its middle, drawn perpendicular to the plane of rotation of the wheel, this axis of symmetry is also the neutral axis.

The tensile stress T_1, due to bending, in the inner fibers of the rim over the arms, in pounds per square inch, is expressed by the formula

$$T_1 = \frac{Hv^2}{144g}\left\{1 + \frac{K}{6}\left[\frac{57.3\,Ry_1}{Ia} - \left(\frac{1}{A} + \frac{Ry_1}{I}\right)\cot a\right]\right\} \qquad (1)$$

The tensile stress T_2, due to bending, in the outer fibers of the rim, half way between arms, in pounds per square inch, is expressed by the formula

$$T_2 = \frac{Hv^2}{144g}\left\{1 + \frac{K}{6}\left[-\frac{57.3\,Ry_2}{Ia} + \left(\frac{Ry_2}{I} - \frac{1}{A}\right)\operatorname{cosec} a\right]\right\} \qquad (2)$$

The tensile stress T_3 in the outer fibers of the arms at the hub, in pounds per square inch, is expressed by the formula

$$T_3 = \frac{Hv^2K}{432gA_1} + \frac{Tc}{144nI} \qquad (3)$$

The tensile stress T_4 in the arms at the rim, in pounds per square inch, is given by the formula

$$T_4 = \frac{Hv^2K}{432gA_2} \qquad (4)$$

The original formulas were derived for stresses in pounds per square foot, but the factor 144 has been introduced in the denominators of the right-hand members of the formulas, so that the stresses are now expressed in pounds per square inch and can be compared directly with other stresses. The units of length and area, however, are expressed in feet and square feet, and should be so substituted in applying the formulas.

The value of K is determined by the formula

$$K = \frac{3 - \left(\frac{r_1 - r_2}{R}\right)^2\left(\frac{r_1 + \frac{1}{2}r_2}{R}\right)}{\frac{1}{A_1}\left(\frac{r_1 - r_2}{R}\right) + \frac{57.3}{2Aa}} \qquad (5)$$

35. If the calculation of the stress by the foregoing methods does not show a large factor of safety, the design must be modified accordingly. This may be done by any one of the following methods or combinations of them:

1. By lowering the linear velocity of the rim. This is the most certain means of affecting the result. As the engine speed cannot be changed, the diameter of the wheel must be reduced. Changing the rim speed involves a complete repetition of all the work of design, beginning with a recalculation of the weight. A lowering of the rim speed also increases the cost of the wheel, as stated in Art. **28.**

2. If the speed has not been assumed at a higher value than 88 feet per second, as given in Art. **28,** it is better to seek increased safety by increasing the *strength* of the wheel, rather than by decreasing the speed. The strength may be increased by any of the following methods: Altering the disposition of the metal in the rim, so as to give the rim greater strength against bending in the plane of rotation. This is often done by putting on ribs extending inwards from the rim. When this is done, it should always be determined by calculation that the modulus of the cross-section, $\frac{I}{c}$, of the rim is really increased, as it is possible to add ribs in such a way as to weaken the rim. A sure means of accomplishing the desired effect is to decrease the width and increase the thickness of the rim, making it hollow if necessary; but if the wheel is to carry a belt, the extent to which the width can be reduced is limited by the width of the belt.

3. By using material of greater tensile strength in proportion to its weight. This method is often successfully employed. A large number of flywheels with wooden rims have been designed and built, and have also been very successful. It is now common to use a steel rim cast in at least two segments separate from the arms, the whole being bolted together in assembling the wheel. Rims are also sometimes built up of boiler plates.

4. By increasing the number and decreasing the size of the arms. Excellent results have been attained by this

method. Band-saw wheels, built like bicycle wheels, with small adjustable pipe arms 6 or 8 inches apart along the rim, are successfully run at 10,000 feet per minute. This is a special construction, and is too expensive for ordinary engine flywheels.

It should be borne in mind that nothing can be done toward increased safety by simply enlarging the dimensions of the rim without changing its shape, material, or construction. Any increase in the weight of the rim increases the centrifugal force that tends to tear the wheel apart, in just the same proportion that it increases the cross-section tending to resist rupture of the rim.

36. Flywheel-Rim Joints.—The rim joints may now be designed. Fig. 19 shows the usual form when the rim joint comes directly over an arm, the rim of the thickness t being broken off at r, r. There will usually be a line of bolts with center lines as at $e f$, and a line of bolts on each side of the joint with center lines as at $h i$ and $h_1 i_1$. At m is shown the outer end of an arm whose center line coincides with the part $d k$ between the two rim sections.

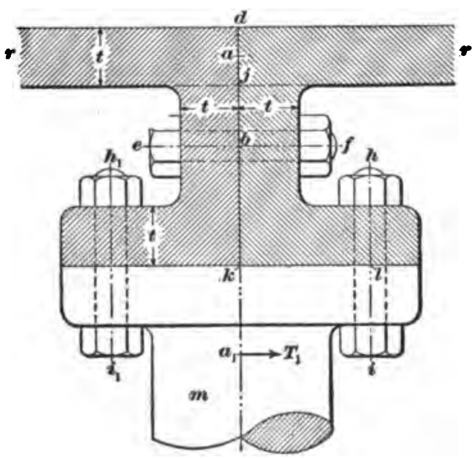


FIG. 19

Let a, Fig. 19, indicate the center of gravity of the cross-section of the rim, not including the flanges for the joint. The stresses in the rim due to bending produce an equivalent tension T_1, which is given by a formula also derived by Professor Lanza. Thus,

$$T_1 = \frac{H v^2}{g}\left(A - \frac{K}{6} \cot a\right) \qquad (1)$$

This tension acts at a point a_1, Fig. 19, such that $a a_1$ is given by the formula

$$a a_1 = \frac{KR\left(\frac{57.3}{a} - \cot a\right)}{6A - K \cot a} \qquad (2)$$

The symbols are the same as those used in Art. **34.**

By taking moments about the point d, Fig. 19, the tension on the bolts at e, f, due to the tension in the rim, is $T_1 \times d a_1 \div d b$. To this must be added the tension on the bolts due to screwing up. This may be estimated as $T_2 = 18{,}700 \times d_1$, where d_1 is the diameter, in inches, of the bolts at the bottom of the threads. The cross-section at the bottom of the threads of the bolts through the joint at $e f$ must be sufficient to carry the total tensile stress on them, as just calculated. The formula for the diameter of bolts then becomes

$$d_1 = \frac{11{,}900}{S_1 n} + \sqrt{\frac{T_1 \times d a_1}{.7854\, d b \times S_1 n} + \left(\frac{11{,}900}{S_1 n}\right)^2}, \qquad (3)$$

in which S_1 = safe tensile stress in bolts, in pounds per square inch;

n = number of bolts in flange at distance $d b$ from outside of rim;

T_1 = equivalent tension, in pounds, acting at a_1;

$d a_1$ and $d b$ = distances on the rim, as shown in Fig. 19, measured in inches.

Ordinarily, the factor of safety in the bolts of the rim joints of existing flywheels is low.

Owing to close competition, designers have sometimes allowed stresses as high as 12,500 pounds per square inch of cross-section in wrought-iron bolts, and 15,000 in steel bolts, but these values are excessive; the stresses should be kept as low as possible.

37. Stress in Rim Flange.—The stress in the rim flange should also be calculated. This stress is composed of a combination of two stresses—that due to the tendency to bend at the point where the flange is attached to the rim, as at j, Fig. 19, owing to the equivalent tension T_1, and that due to the centrifugal tension, as calculated by formula **4**, Art. **34**.

The bending moment of T_1 about the point j is $T_1 \times j a_1$, and the moment of resistance of the section of the flange at j is $\frac{S w t^2}{6}$, which gives the stress due to bending as

$$S = \frac{6 T_1 \times \overline{j a_1}}{w t^2}, \qquad (1)$$

in which T_1 = total tension at a_1, as calculated by formula **1**, Art. **36**;
w = width of flange, in inches;
t = thickness of flange, in inches;
S = stress in flange due to bending.

Then, adding this stress to the tensile stress due to the centrifugal force, as found by formula **4**, Art. **34**, gives the total stress S_1 as

$$S_1 = \frac{6 T_1 \times \overline{j a_1}}{w t^2} + \frac{H v^2 K}{6 g w t} \qquad (2)$$

in which H, v, g, and K are as given in Art. **34**, and the other symbols are the same as just stated. It will be noticed that $w t$, in square inches, appears in the last part of the formula in place of the area $144 A_1$ of cross-section of the arm, and as there are two flanges of thickness t to take the stress due to centrifugal force, the denominator is $6 g w t$ instead of $432 g A_1$.

At first, the thickness of rim and flange should be made the same, to improve the chances of a good casting; then, S_1 should be calculated by the foregoing formula. If S_1 should appear excessively large, the thickness of the flange should be increased, until S_1 is reduced to a safe value.

38. Stress in Bolts Fastening Arm to Rim.—The bolts fastening the rim flange to the arms are located as shown at $h i$ and $h_1 i_1$, Fig. 19. The total stress in these bolts is the sum of the stress due to the centrifugal force, that due to screwing up the nuts, and that due to the tension T_1 in the rim caused by bending. T_1 tends to produce a rotation of the flange about the point d, and the moment is

$T_1 \times da_1$. This moment divided by kl, the perpendicular distance from hi to d, gives the tension on the bolts. The tension due to screwing up the nuts may be taken, as in Art. **36,** as $T_2 = 18{,}700\,d_1$. The tension due to the centrifugal force is $\frac{Hv^2K}{6g}$, which is $72A_2$ times the value of the tensile stress, in pounds per square inch, as given in formula **4,** Art. **34,** there being two sections and the total tension on each being considered. Hence, the total tension on these bolts is found by taking the sum of these three tensions, as shown by the formula

$$T = \frac{T_1 \times \overline{da_1}}{\overline{kl}} + \frac{Hv^2K}{6g} + 18{,}700\,d_1 \qquad (1)$$

The total tension as thus found should equal the product of the area of cross-section of half the bolts on one arm and the safe stress. This gives a formula for the diameter that reduces to $d_1 =$

$$\frac{11{,}900}{S_1 n} + \sqrt{\frac{T_1\,\overline{da_1}}{.7854\,\overline{kl} \times S_1 n} + \frac{Hv^2K}{4.7124\,S_1 ng} + \left(\frac{11{,}900}{S_1 n}\right)^2}, \qquad (2)$$

in which S_1 may have values as given in Art. **36,** and n is the number of bolts fastening one section of the rim to an arm, or half the total number of bolts fastening the rim to one arm.

39. Strength of Joint Between Arms.—When a built-up flywheel is made so that the joint comes between the arms, it should be located at a distance from the center line of one arm equal to $.211\,x$, where x represents the distance between the middle lines of adjacent arms, measured along the middle of the rim. Fig. 20 represents a joint located between arms as just stated. The rim is broken away at a, a, and the tension T in the rim acts at the point a_1, which may be taken, without serious error, as the center of gravity of

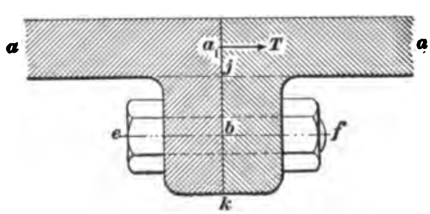


FIG. 20

the rim section, neglecting the flanges. The total tension on one joint located as just stated may be found by using the following formula, which has been derived by means of higher mathematics:

$$T = \frac{Hv^2}{g}\left[A - \frac{K}{6}\,\frac{\cos\left(a - \frac{12.1\,x}{R}\right)}{\sin a}\right], \qquad (1)$$

in which the symbols are the same as explained in Art. **34.** R and x, of course, must be in the same units.

Then, as the tension of the bolts acts at b and the flange in opening would turn about the point k, the tension T_1 in the bolts may be found by using the formula

$$T_1 = \frac{T \times \overline{a_1 k}}{\overline{b\,k}}. \qquad (2)$$

It is apparent that the stress T_1 in the bolts is diminished by putting them as close to the rim as possible; that is, by increasing $b\,k$. The tension T_2 due to screwing up may be estimated as being equal to 18,700 d, when d represents the diameter of the bolts in inches at the root of the thread. The cross-sectional area of the bolts at the bottom of the threads must be made sufficient to carry the total tension without excessive stress. Placing the product of the area of cross-section of a bolt, the number of bolts, and the stress equal to the sum of these tensions, gives the expression

$$.7854\;d^2 n S_1 = \frac{T \times \overline{a_1 k}}{\overline{b\,k}} + 18{,}700\,d$$

This reduces to the formula

$$d = \frac{11{,}900}{n S_1} + \sqrt{\frac{T \times \overline{a_1 k}}{.7854\,\overline{b\,k} \times S_1 n} + \left(\frac{11{,}900}{n S_1}\right)^2} \qquad (3)$$

40. Besides being subjected to tension, the bolts at e, f, Fig. 20, are subjected to a shearing force that may be calculated by the use of the following formula, which has been derived by means of higher mathematics:

$$S = \frac{H v^2 K}{6g}\left[\frac{\sin\left(\frac{a - 12.1\,x}{R}\right)}{\sin a}\right],$$

in which the symbols have the same meaning as in Art. **34.** R and x, of course, must be taken in the same units. The shear S is the total shear on all the bolts in one joint, and to get the unit stress, S must be divided by the area of cross-section of all the bolts in that joint. The shearing stress should not exceed 10,000 pounds per square inch for wrought iron nor 12,500 for steel, and, if possible, should be kept at a lower figure. The shearing stress is carried at the joint, so that the entire area of cross-section of the bolts, rather than that at root of threads, should be used here. The thickness of the flange may be made the same as that of the rim.

41. Flywheel Hub.—When the hub of a flywheel is cast in sections, it must be fastened together with fastenings designed to resist the forces that tend to rupture the hub. There are two common methods of securing together the parts of the hub—one is by shrinking wrought-iron or steel bands around the hub, and the other is by bolting. In the absence of more accurate information, it will be safe to assume that the force tending to separate the halves of the hub is equal to that which tends to tear the rim apart.

Consider first the method of fastening the hub by means of bands. From formula **4**, Art. **27**, the total force that tends to separate the rim at any section is $\frac{H A v^2}{4{,}631}$. If this is resisted by the rings, the area of the sections of the rings must be sufficient, so that the stress shall not be excessive. The product of the area of cross-section of one ring, the number of rings, and the stress must equal $\frac{H A v^2}{4{,}631}$; that is,

$$A_2 m S_1 = \frac{H A v^2}{4{,}631},$$

in which A_2 = area of section of one band, in square inches;
S_1 = safe tensile stress, in pounds per square inch;
m = number of bands, which will generally be two;
H = weight of metal in rim, in pounds per cubic foot;
A = area of cross-section of rim, in square inches;
v = velocity of rim, in feet per second;
g = 32.16.

Solving the foregoing expression for A_2 gives the formula

$$A_2 = \frac{H A v^2}{4{,}631\ m\ S_1} \qquad (1)$$

S_1 should not exceed from 1,500 to 3,000. The reason for limiting the calculated stress to this low value is that there is a much greater and uncalculated stress in the bands due to shrinking them on. In a cast-iron wheel, $H = 450$, and formula **1** becomes

$$A_2 = .09717 \frac{A v^2}{m\ S_1} \qquad (2)$$

42. In a similar manner, if the sections of the hub are to be bolted together, the bolts must be designed to resist similar stresses. In fact, the formulas of Art. **41** may be used directly for the area of cross-section of the bolts in a joint on one side of the hub. In such a case, m would be the number of bolts on one side of the hub, and A_2 the area of cross-section of one bolt.

In the case of bolts, however, there are no shrinkage stresses, but there is the stress due to screwing up the nut. The tensile stress in the bolts should not exceed 2,500 pounds per square inch of cross-section at the root of the threads.

The hub should be of good substantial dimensions, from about 2 inches thick in the smallest wheel to about 6 inches thick in the largest, and long enough to get a good solid bearing on the shaft. The hub contains an excess of metal for all calculable stresses on it, and it is not necessary to make any calculations regarding them.

43. General Construction of Flywheels.—As has already been stated, flywheels of small diameter are usually cast solid; the arms are of elliptical cross-section, and the wheel has the general appearance of a belt pulley with a heavy rectangular rim. Large flywheels, however, are cast in sections or built up of plates and castings.

A heavy flywheel with solid, elliptical arms is shown in Fig. 21. This wheel is cast in four sections, with two arms to each section, and the hub is formed of two separate rings, which are bolted and keyed to the segments forming the inner ends of each pair of arms. The segments of the rim are joined by means of steel or wrought-iron rings *R*, which are shrunk on bosses formed by recesses cast in the rim.

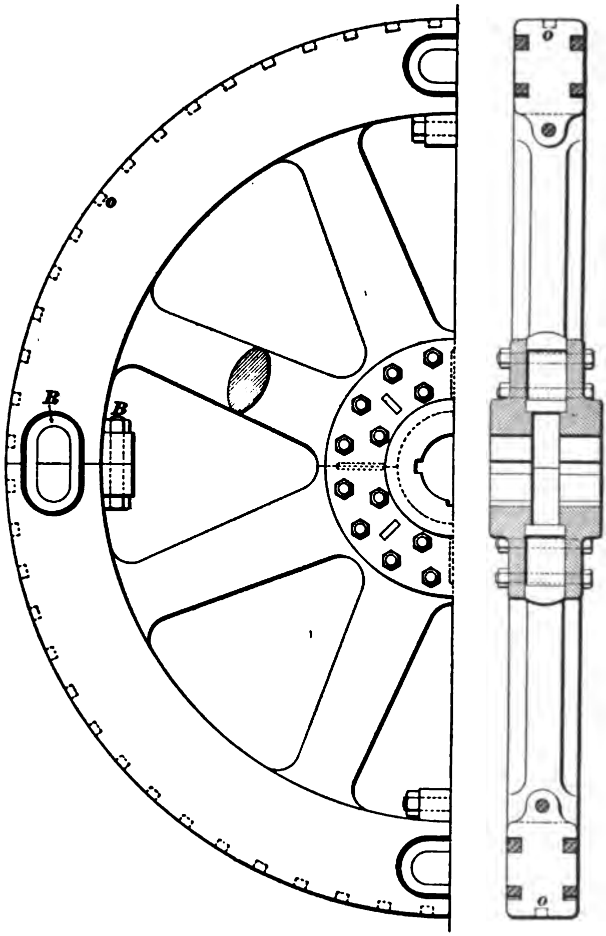


Fig. 21

Besides these rings, bolts *B* pass through lugs on the inner surface of the rim. In order that the wheel may be amply strong, the net section of the bolt and two rings must be sufficient to withstand the total tension due to centrifugal force tending to separate the rim through the section that they join. This tension may be calculated by formula **4**, Art. **27**.

The bolts and keys joining the inner ends of the arms to

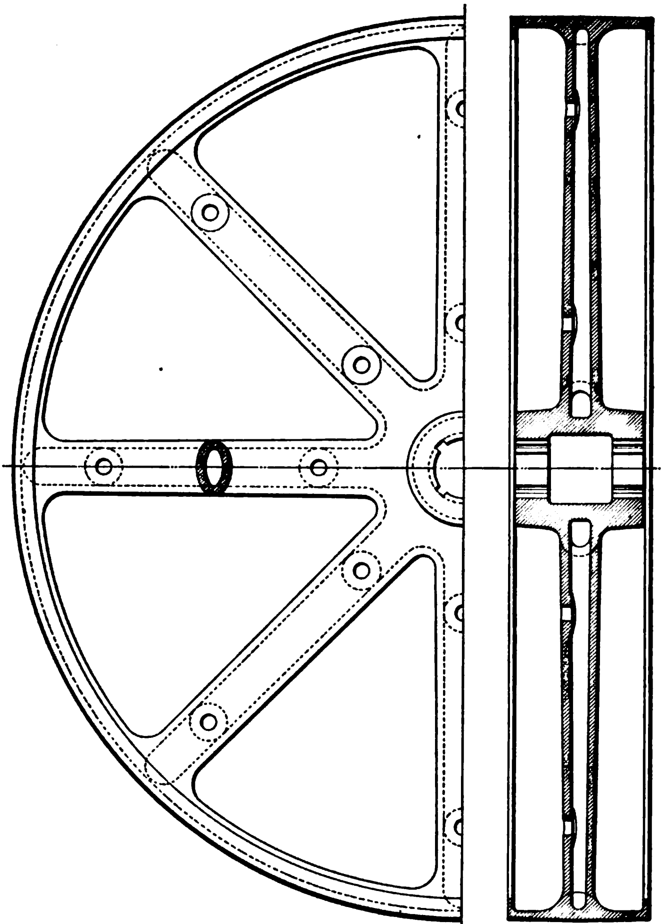


FIG. 22

the hub are in double shear, and they must be calculated to withstand the pull exerted by the rim on each arm. This pull, as derived from formula **4**, Art. **34**, is

$$F = \frac{H v^2 K}{3 g}$$

44. Fig. 22 shows a flywheel with the face of the rim turned to serve as a belt pulley. The arms are oval in section and cast hollow, thus giving them increased stiffness for a given weight. The rim is also given a channel-shaped section, which increases its ability to withstand the bending stresses produced by centrifugal force in the sections between the arms.

45. Another method of fastening the arms to the hub is shown in Fig. 23. Here, the ends of the arms are cast with

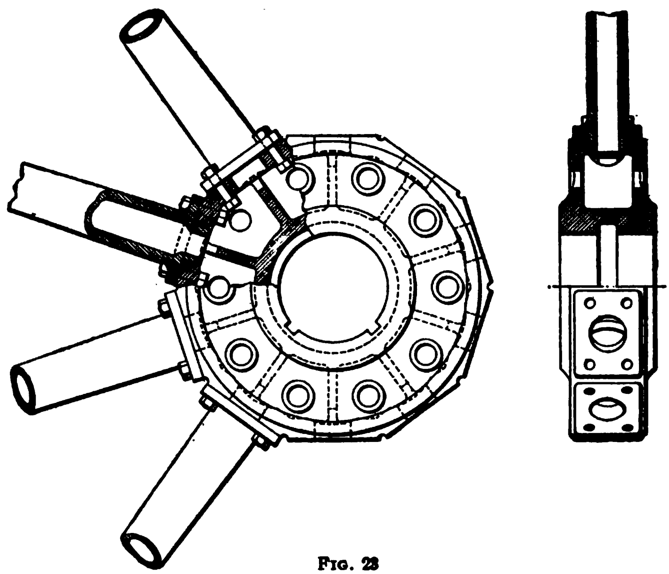


Fig. 23

flanges, and circular bosses, which fit into recesses bored in the hub, are also turned on them. Bolts pass through the flanges in the ends of the arms and in the face of the hub. thus holding the arms securely in place.

46. Fig. 24 shows a flywheel cast in halves. The sections of the rim are joined by means of steel or wrought-iron bars *b* inserted in holes cast in the ends of the rims. These bars are fastened to the rim by keys *k* that pass through holes fitted for that purpose. The arms are oval in section and are cast solid. The hub is provided with bosses through which four bolts are passed, thus joining the two parts of the hub securely. The holes *o* cored in the rim of the flywheels shown in Figs. 21 and 24 are for the purpose of inserting

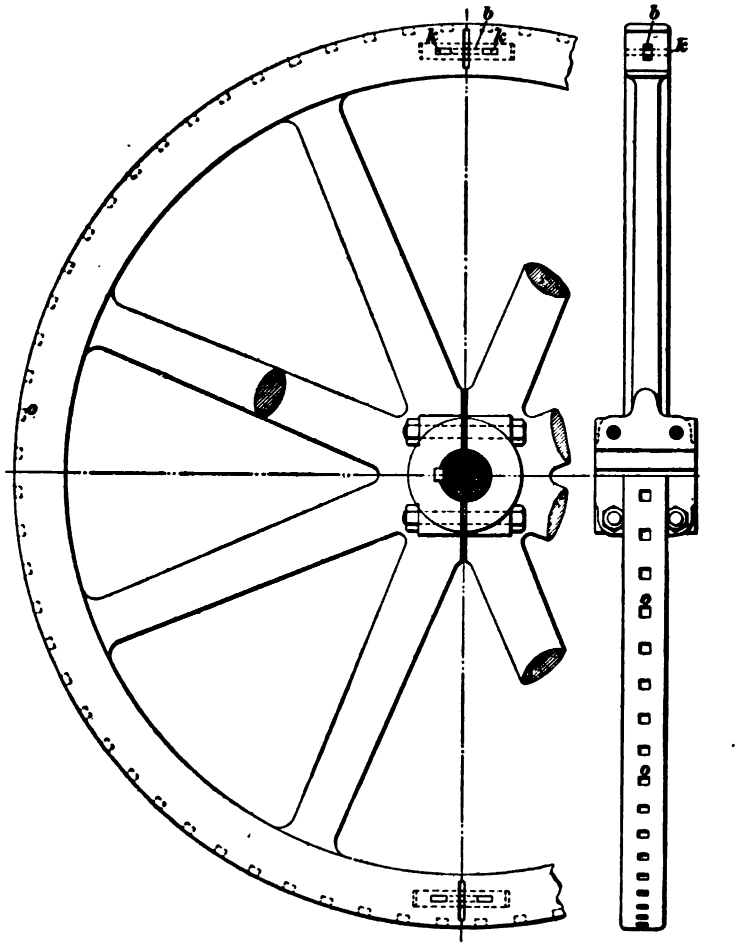


Fig. 24

the end of a bar when it is required to turn the engine, either to get the crank off the dead center in starting or for any other purpose.

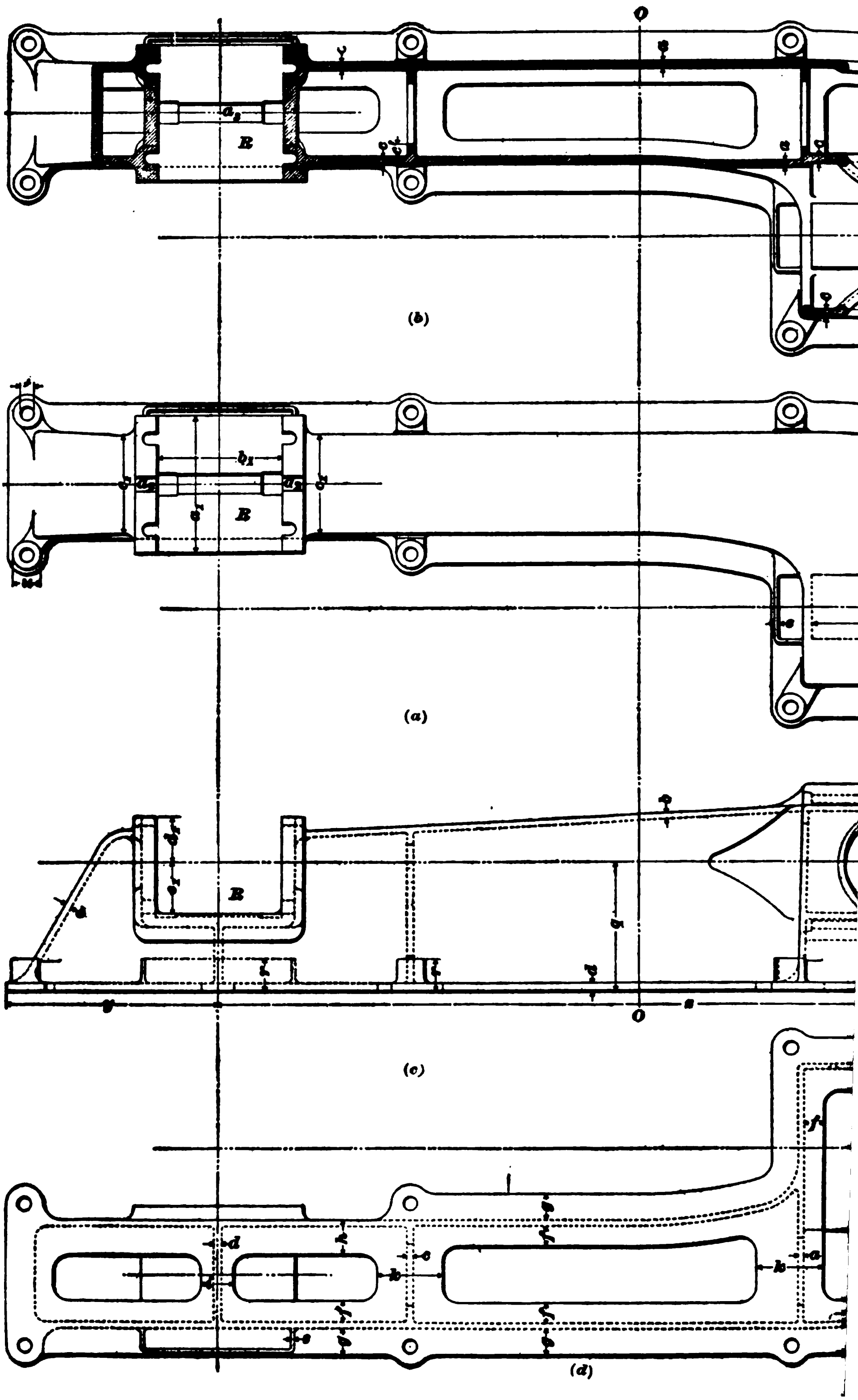
ENGINE FRAMES, OR BEDS

47. The **frame,** or **bed,** of an engine is the main structure to which the other parts are attached. It is stiff and rigid, and, on account of its mass, absorbs more or less of the vibration due to the movement of the reciprocating parts. Engine beds are made in a great variety of forms, each type of engine having its peculiar type of bed.

Fig. 25 shows a substantial type of engine bed for horizontal engines. Fig. 25 (*a*) is a top view, (*b*) a horizontal section on the center line looking toward the bottom, (*c*) a front side view, (*d*) a view of the bottom, (*e*) a cross-section through the center line of the main bearing, (*f*) a cross-section on the line *O O*, (*g*) an end view, and (*h*) a cross-section through the guides on the line *P P*. The guides *L*, *L* are cast solid with the bed, and then bored out to form the bearing surface for the crosshead, which is of the form shown in Fig. 1. *I*, *I* are bosses, which form bearings for the rocker-arm shaft. These bearings are provided with brass or Babbitt bushings. The main bearing, which is separate from the frame, rests in the opening *R*.

Proportions for designing this bed are based on the diameter of the cylinder, the length of stroke, dimensions of crosshead, and length of connecting-rod, as follows:

D = diameter of cylinder.
$a = .027\,D + .1875$ inch.
$b = 1.1\,a$.
$c = 1.25\,a$.
$c' = 2.5\,a$.
$d = 1.5\,a$.
$e = 2.25\,a$.
$f = 4\,a$.
$g = 4.75\,a$, but never less than u.
$h = 6.5\,a$.
$i = 6\,a$.
$k = 12\,a$.
$k' = 13.5\,a$.
$l = 2\,a$.
m = the distance required to clear the connecting-rod.
$n = 2\,a$.
$o = .5\,a$.
$p = .75\,m$.
$q = .8\,D$ to D.
$r = 6\,a$.
$s = .5\,a$.
$t = .06\,D + .5$ inch (use the nearest standard size of bolt).
$u = 2.1\,t$.


(b)
(a)
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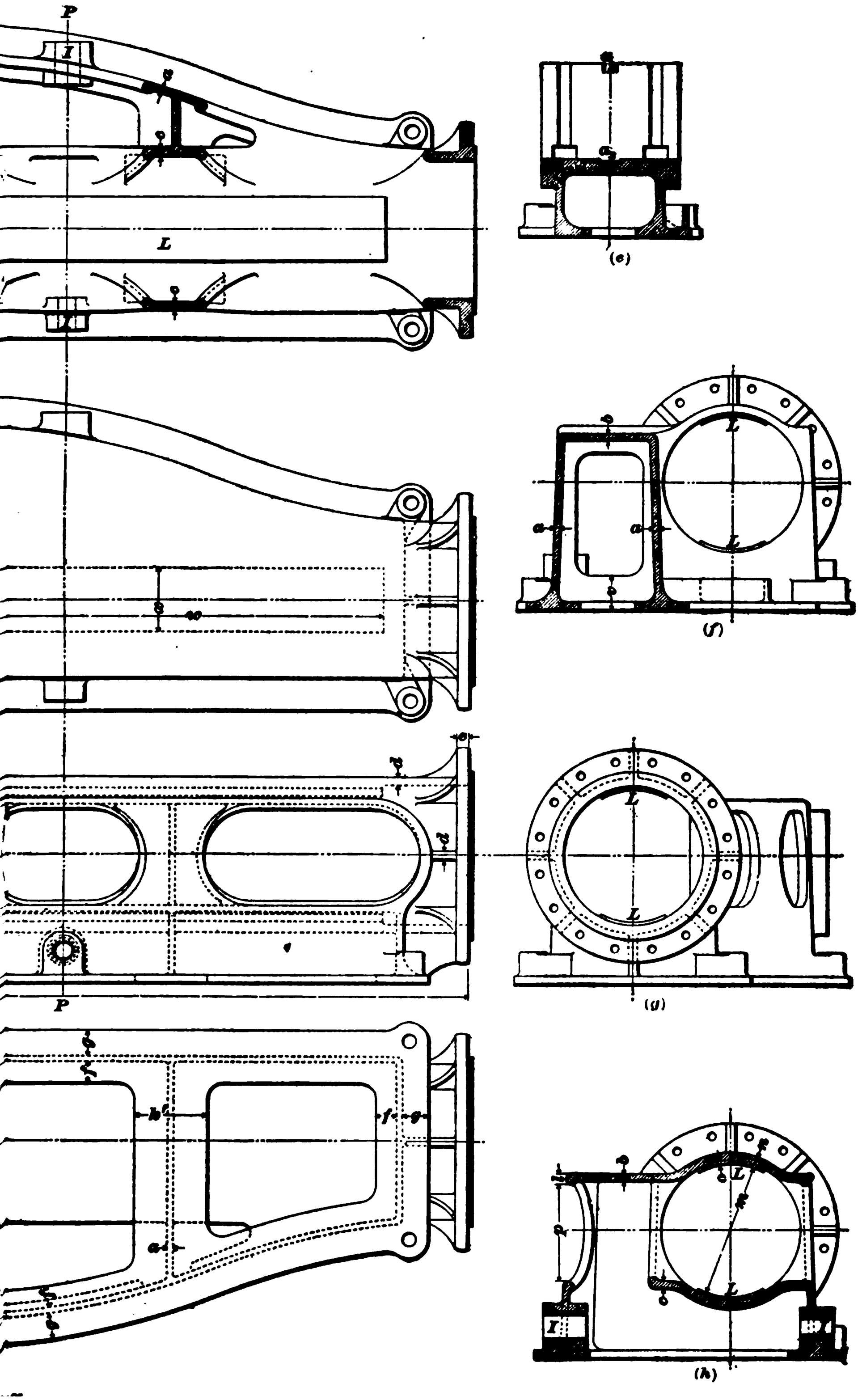
(d)



P
L
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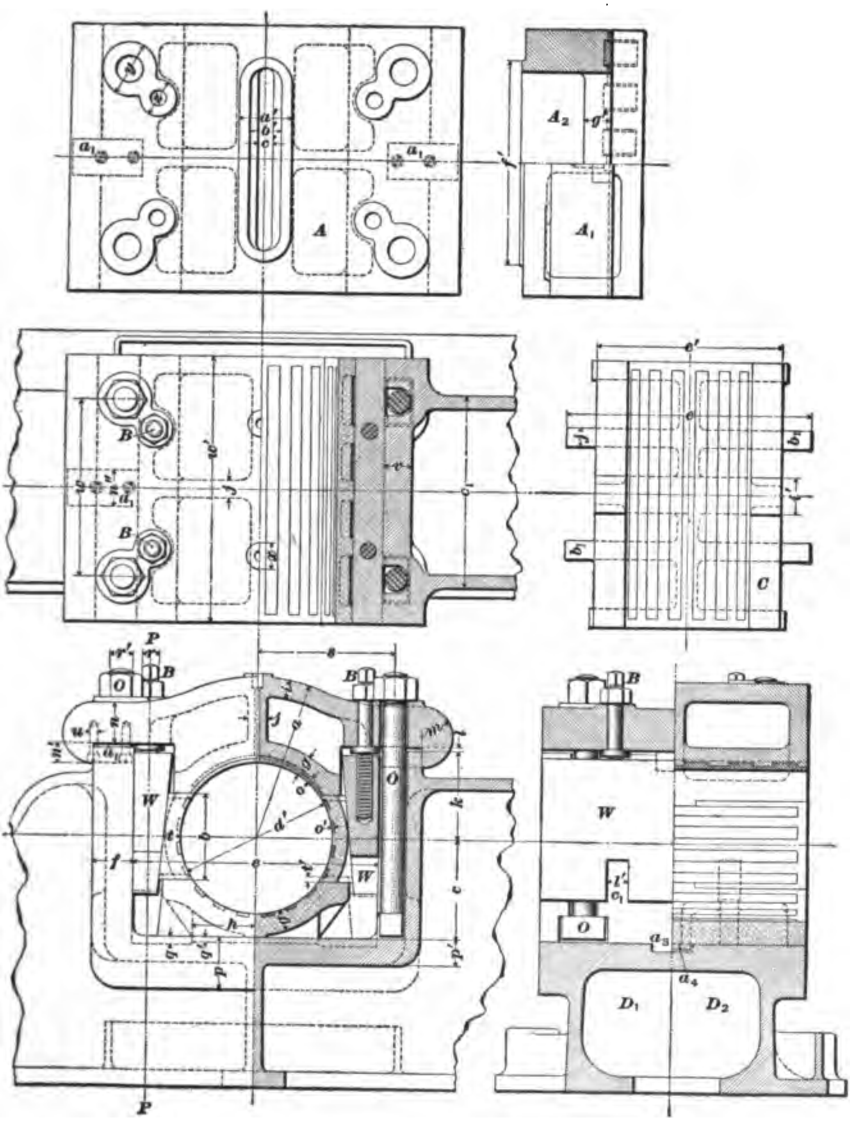


51

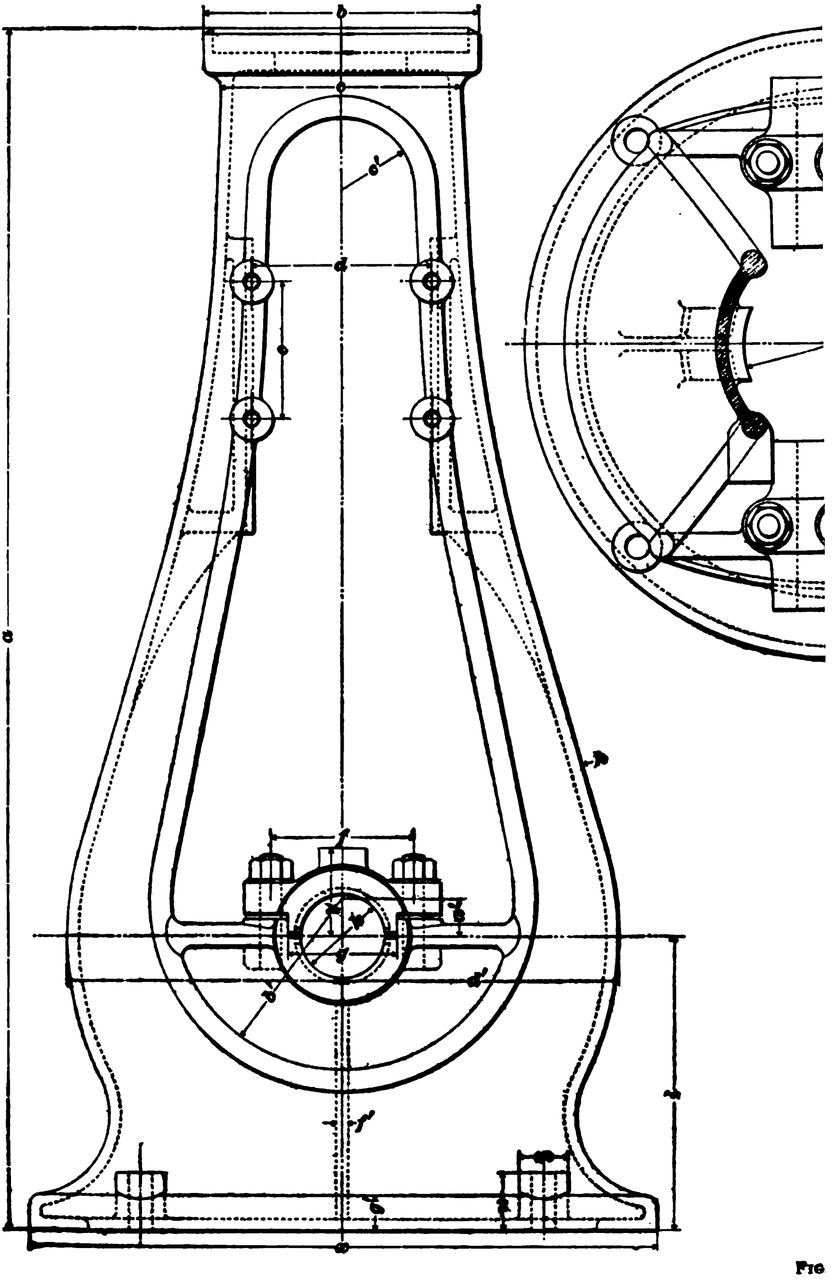
FIG. 26

v = 6.5 a.
w = length of stroke + length of rubbing surface of crosshead − (.01 D + .1875 inch).
x = same width as crosshead.
y = about 1.3 D. In all cases, the crank must clear the bosses and nuts for the foundation bolts.

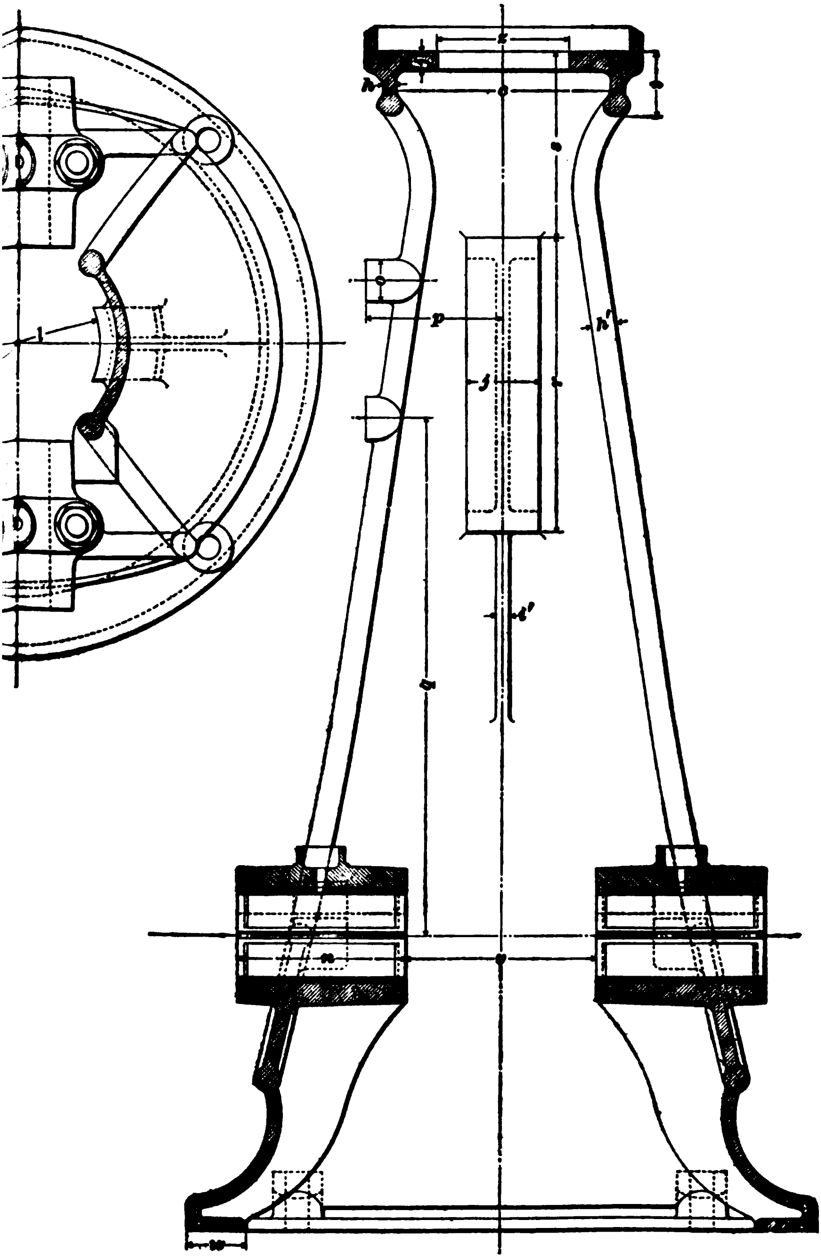
The length z must be such that the hub of the crosshead will clear the stuffingbox bolts when at the end of the stroke. Approximately, its value = length of crank + length of connecting-rod + distance from center of crosshead pin to end of crosshead hub + clearance between crosshead hub and stuffingbox bolts + the distance that the stuffingbox bolts project into the frame. This distance z is best determined by laying out the various parts to scale.

The dimensions for the seat for the main bearing are:
d' = diameter of crank-shaft journal.
a_1 = 1.75 d'.
b_1 = 1.65 d' − .5 inch.
c_1 = .62 D.
d_1 = .5 d' + 1.25 inches.
e_1 = .66 d'.

The bearing for the frame shown in Fig. 25 is shown in detail in Fig. 26. Proportions for designing this bearing are:
d' = diameter of journal.
D = diameter of cylinder.
a = d' + 1 inch.
a' = .2 d' + 2 inches.
b = .5 d' + 1 inch.
b' = .12 d' + 1.25 inches.
c = .66 d'.
c' = .06 d' + .625 inch.
c_1 = .62 D.
e = 1.65 d' − .5 inch.
e' = 1.22 d'.
f = .25 d' + .375 inch.
f' = 1.35 d'.
g = .1 d' + .5625 inch.
g' = .1 d' + 1 inch.
h = .85 d'.
i = .1 d' + .25 inch.
i' = .2 d' + .5 inch.
j = .1 d' + .25 inch.
k = .5 d' + 1.25 inches.
l = .375 inch, constant.
l' = .1 d' + .375 inch.
m = .175 d' + .3125 inch.
n = .25 d' + .25 inch.
n' = .1 d' + .375 inch.
n'' = .2 d' + .5 inch.



FIG



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$o = .625$ inch, constant.
$o' = .375$ inch, constant.
$p = .3\,d' + .5$ inch.
$p' = .15\,d' + .375$ inch.
$q = .02\,d' + .5$ inch.
$q' = .02\,d' + .25$ inch.
$r = .1\,d'$.
$r' = .15\,d'$.
$s = .9\,d'$.
$t = .11\,d'$.
$t' = .02\,d + .25$ inch.
$u = .04\,d' + .125$ inch; use nearest standard size bolt.
$v = .15\,d' + .375$ inch.
$w = 1.2\,d'$.
$w' = 1.75\,d'$.
$x = 2.5$ inches, constant.
$y = .3\,d' + .75$ inch.
$z = .2\,d' + .5$ inch.

It will be observed that the bearing shown in Fig. 26 has four seats, including the cap, which is lined with Babbitt so as to form the top seat. The two side seats can be adjusted by means of the wedges *W*, which are moved by the bolts *B*. The bearing is held to the engine bed by the **T**-head bolts *O*, *O*, which fit into slots cast in the bed for this purpose. The side and bottom seats are of brass, with Babbitt linings. *A* is a top view of the cap, A_1 a side view of the cap, and A_2 a section of the cap on the center line. Lugs a_1, usually of wrought iron, fit into slots a_2, Fig. 25, to prevent end motion of the cap. *C*, Fig. 26, is an inside, or top, view of the bottom seat. It has lugs b_1 that fit into the slots b_2 of the wedges *W* (see view D_1, which is a half section through the bearing on the line *PP*, with the wedge in place). D_2 is a half section of the bearing on the center line. The bottom seat has a lug a_4 (see section D_2) that fits into a corresponding slot in the bed. This slot is shown at a_3 in section D_1; also at a_3, Fig. 25.

48. Fig. 27 is an example of a frame for a vertical engine, as made by a well-known builder. The dimensions of this frame for various sizes of cylinders are given in Table I.

TABLE I

DIMENSIONS OF FRAME FOR VERTICAL ENGINE

Size of Engine Inches	*a*	*b*	*c*	*d*	*e*	*f*	*g*	*h*	*i*	*j*	*k*	*l*	*m*
3 × 5	$28\frac{1}{4}$	$6\frac{7}{8}$	6	$3\frac{3}{4}$	$3\frac{1}{4}$	$3\frac{5}{8}$	$2\frac{1}{4}$	$\frac{1}{4}$	$6\frac{3}{4}$	$1\frac{1}{4}$	$1\frac{3}{4}$	$4\frac{1}{2}$	$1\frac{1}{2}$
4 × 6	$34\frac{3}{4}$	$7\frac{7}{8}$	$7\frac{1}{4}$	$5\frac{1}{4}$	$4\frac{1}{2}$	$4\frac{1}{2}$	3	$\frac{3}{8}$	$7\frac{3}{4}$	$1\frac{3}{4}$	2	5	$1\frac{1}{2}$
$5\frac{1}{2}$ × 7	$40\frac{1}{4}$	$9\frac{1}{2}$	8	6	$5\frac{1}{2}$	5	$3\frac{5}{8}$	$\frac{3}{8}$	10	$2\frac{1}{2}$	$2\frac{1}{2}$	6	$1\frac{3}{4}$
7 × 9	$50\frac{1}{4}$	$11\frac{1}{2}$	10	$7\frac{1}{2}$	$5\frac{3}{4}$	$6\frac{1}{4}$	4	$\frac{1}{2}$	$12\frac{3}{8}$	$2\frac{1}{2}$	$2\frac{15}{16}$	$7\frac{7}{8}$	2
9 × 12	$61\frac{1}{4}$	$14\frac{1}{4}$	$12\frac{1}{2}$	9	$7\frac{1}{4}$	9	$6\frac{3}{4}$	$\frac{5}{8}$	$13\frac{3}{4}$	$3\frac{1}{2}$	$3\frac{15}{16}$	$9\frac{1}{2}$	2

Size of Engine Inches	*n*	*o*	*p*	*q*	*r*	*s*	*t*	*u*	*v*	*w*	*x*	*y*	*z*
3 × 5	$3\frac{3}{8}$	1	$3\frac{1}{16}$	$12\frac{3}{8}$	$7\frac{3}{4}$	$2\frac{15}{16}$	$1\frac{3}{4}$	$1\frac{5}{8}$	$3\frac{3}{8}$	$\frac{1}{2}$	$12\frac{3}{4}$	2	$3\frac{5}{8}$
4 × 6	$4\frac{7}{16}$	$1\frac{1}{2}$	$3\frac{7}{8}$	$13\frac{3}{4}$	$9\frac{3}{8}$	$4\frac{3}{4}$	$2\frac{5}{8}$	$1\frac{5}{8}$	$4\frac{1}{8}$	$\frac{5}{8}$	15	$2\frac{1}{4}$	$4\frac{1}{2}$
$5\frac{1}{2}$ × 7	$5\frac{13}{16}$	$1\frac{1}{2}$	$4\frac{3}{8}$	$15\frac{1}{2}$	$10\frac{1}{2}$	$4\frac{3}{4}$	$2\frac{1}{4}$	$2\frac{5}{8}$	$4\frac{7}{8}$	1	18	$3\frac{1}{2}$	$4\frac{3}{4}$
7 × 9	$5\frac{5}{8}$	$1\frac{3}{4}$	5	$21\frac{5}{8}$	$12\frac{1}{4}$	$7\frac{3}{4}$	$3\frac{3}{4}$	3	$6\frac{3}{4}$	$2\frac{1}{2}$	$26\frac{1}{4}$	$4\frac{1}{4}$	$5\frac{1}{2}$
9 × 12	$5\frac{1}{2}$	2	$5\frac{3}{4}$	$30\frac{1}{8}$	$17\frac{1}{2}$	$7\frac{1}{2}$	$4\frac{3}{4}$	$3\frac{1}{2}$	11	$3\frac{1}{4}$	$26\frac{1}{2}$	6	$7\frac{1}{2}$

Size of Engine Inches	*a'*	*b'*	*c'*	*d'*	*e'*	*f'*	*g'*	*h'*	*i'*	*j'*
3 × 5	$\frac{11}{16}$	4	$2\frac{3}{16}$	$11\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{8}$	1	$\frac{1}{2}$	$\frac{3}{8}$	$\frac{1}{2}$
4 × 6	$\frac{15}{16}$	$4\frac{5}{8}$	$2\frac{1}{2}$	$13\frac{1}{2}$	$\frac{5}{8}$	$\frac{1}{2}$	$1\frac{1}{8}$	$\frac{3}{4}$	$\frac{1}{2}$	$\frac{3}{4}$
$5\frac{1}{2}$ × 7	1	$5\frac{1}{4}$	$2\frac{5}{8}$	$15\frac{3}{4}$	$\frac{3}{4}$	$\frac{1}{2}$	$1\frac{1}{4}$	$\frac{3}{4}$	$\frac{1}{2}$	$\frac{3}{4}$
7 × 9	$2\frac{3}{16}$	$7\frac{7}{8}$	$3\frac{11}{16}$	23	$\frac{7}{8}$	$\frac{9}{16}$	$1\frac{1}{2}$	1	$\frac{9}{16}$	1
9 × 12	$\frac{3}{4}$	$7\frac{5}{8}$	$4\frac{7}{16}$	$23\frac{1}{2}$	$\frac{7}{8}$	$\frac{5}{8}$	$1\frac{3}{4}$	$1\frac{1}{4}$	$\frac{5}{8}$	$1\frac{1}{4}$

e' = size of foundation bolt. For 9″ × 12″ engine, a disk center crank is used.

EXAMPLES OF ENGINE PROPORTIONS

49. It has been previously stated that when a standard line of engines is to be manufactured, the rules and formulas

TABLE II

PROPORTIONS OF CORLISS ENGINES

Diameter of Cylinder Inches	Stroke Inches	Revolutions per Minute	Piston Speed Feet per Minute	Diameter of Shaft Journal Inches
10	24	90	360	5
12	24	90	360	$5\frac{1}{2}$
14	30	80	400	7
16	30	80	400	$7\frac{1}{2}$
16	36	70	420	8
18	36	70	420	$8\frac{1}{2}$
20	36	70	420	9
16	42	65	455	$8\frac{1}{4}$
18	42	65	455	$8\frac{3}{4}$
20	42	65	455	$9\frac{1}{4}$
20	48	60	480	$9\frac{1}{2}$
22	48	60	480	10
24	48	60	480	10
22	54	60	540	$10\frac{1}{4}$
24	54	60	540	$10\frac{3}{4}$
26	54	60	540	$11\frac{1}{4}$
24	60	60	600	11
26	60	60	600	$11\frac{1}{2}$
28	60	60	600	12
30	60	60	600	$12\frac{1}{2}$
32	60	60	600	13
34	60	60	600	$13\frac{1}{2}$
36	60	60	600	14

for the design of the various parts need not be applied to each individual engine. It is found that under the conditions in which the engines are to work, a certain ratio may be assumed to exist between the sizes of the parts. For example, a certain line of engines work uniformly at a steam pressure of 75 pounds, and the length of the piston rod bears a fixed relation to the length of stroke. Under these circumstances, the diameter of the piston rod may be a fixed fraction of the diameter of the cylinder for all sizes, and it is

TABLE III

PROPORTIONS OF CORLISS ENGINES

Diameter of Cylinder Inches	Diam. and Length of Crank-pin Inches	Diameter and Length of Cross-head Pin Inches	Diameter of Valve Stem Inches	Depth of Piston Inches	Diameter of Piston Rod Inches	Width of Crank-Disk Inches	Clearance of Piston Inch
10	$2\frac{1}{2}$	$1\frac{5}{8} \times 2\frac{1}{2}$	$\frac{7}{8}$	4	$1\frac{5}{8}$	$2\frac{1}{2}$	$\frac{1}{4}$
12	3	2×3	1	$4\frac{1}{2}$	2	3	$\frac{1}{4}$
14	$3\frac{1}{2}$	$2\frac{3}{8} \times 3\frac{1}{2}$	$1\frac{1}{8}$	5	$2\frac{1}{4}$	$3\frac{1}{2}$	$\frac{1}{4}$
16	4	$2\frac{5}{8} \times 4$	$1\frac{1}{4}$	$5\frac{1}{2}$	$2\frac{1}{2}$	4	$\frac{1}{4}$
18	$4\frac{1}{2}$	$3 \times 4\frac{1}{2}$	$1\frac{3}{8}$	6	$2\frac{7}{8}$	$4\frac{1}{2}$	$\frac{1}{4}$
20	5	$3\frac{3}{8} \times 5$	$1\frac{1}{2}$	$6\frac{1}{2}$	$3\frac{1}{4}$	5	$\frac{1}{4}$
22	$5\frac{1}{2}$	$3\frac{5}{8} \times 5\frac{1}{2}$	$1\frac{1}{2}$	7	$3\frac{1}{2}$	$5\frac{1}{2}$	$\frac{5}{16}$
24	6	4×6	$1\frac{5}{8}$	$7\frac{1}{2}$	$3\frac{7}{8}$	6	$\frac{5}{16}$
26	$6\frac{1}{2}$	$4\frac{1}{4} \times 6\frac{1}{2}$	$1\frac{3}{4}$	$7\frac{3}{4}$	$4\frac{1}{8}$	$6\frac{1}{2}$	$\frac{5}{16}$
28	7	$4\frac{5}{8} \times 7$	$1\frac{3}{4}$	$8\frac{1}{4}$	$4\frac{1}{2}$	7	$\frac{5}{16}$
30	$7\frac{1}{2}$	$5 \times 7\frac{1}{2}$	$1\frac{7}{8}$	$8\frac{1}{2}$	$4\frac{7}{8}$	$7\frac{1}{2}$	$\frac{5}{16}$
32	8	$5\frac{1}{4} \times 8$	$1\frac{7}{8}$	$8\frac{1}{2}$	$5\frac{1}{8}$	8	$\frac{5}{16}$
34	$8\frac{1}{2}$	$5\frac{5}{8} \times 8\frac{1}{2}$	2	$8\frac{3}{4}$	$5\frac{1}{2}$	$8\frac{1}{2}$	$\frac{3}{8}$
36	9	$5\frac{7}{8} \times 9$	2	9	$5\frac{3}{4}$	9	$\frac{3}{8}$

only necessary, therefore, to multiply the cylinder diameter by this fraction to find the diameter of the piston rod.

Tables II to V, inclusive, give the proportions of a standard line of Corliss engines made by a leading manufacturer. They will serve to illustrate the use of fixed proportions in designing, and will also furnish valuable examples of good, modern practice.

TABLE IV

PROPORTIONS OF CORLISS ENGINES

Diameter of Cylinder Inches	Steam Inlet Area Square Inches	Exhaust Area Square Inches	Diameter of Valve Inches	Inlet Ports Inches	Exhaust Ports Inches	Width of Steam Chest Inches	Width of Exhaust Chest Inches	Depth of Steam and Exhaust Chests Inches	Diameter of Steam Pipe Inches	Diameter of Exhaust Pipe Inches
10	4.71	7.85	$3\frac{1}{8}$	$\frac{1}{2} \times 10$	$\frac{13}{16} \times 10$	5	$6\frac{1}{2}$	$1\frac{3}{4}$	$2\frac{1}{2}$	$3\frac{1}{2}$
12	6.79	11.31	$3\frac{1}{2}$	$\frac{9}{16} \times 12$	$\frac{15}{16} \times 12$	6	$7\frac{1}{2}$	2	3	4
14	9.23	15.39	$3\frac{7}{8}$	$\frac{11}{16} \times 14$	$1\frac{1}{8} \times 14$	7	9	$2\frac{1}{4}$	$3\frac{1}{2}$	$4\frac{1}{2}$
16	12.00	20.00	$4\frac{1}{4}$	$\frac{3}{4} \times 16$	$1\frac{1}{4} \times 16$	8	10	$2\frac{1}{2}$	4	5
18	15.27	25.44	$4\frac{5}{8}$	$\frac{7}{8} \times 17$	$1\frac{1}{2} \times 17$	9	$11\frac{1}{2}$	$2\frac{3}{4}$	$4\frac{1}{2}$	6
20	18.85	31.42	5	1×19	$1\frac{11}{16} \times 19$	10	$12\frac{1}{2}$	3	5	$6\frac{1}{2}$
22	22.80	38.00	$5\frac{3}{8}$	$1\frac{1}{8} \times 20\frac{7}{8}$	$1\frac{7}{8} \times 20\frac{7}{8}$	11	$14\frac{1}{2}$	$3\frac{1}{4}$	$5\frac{1}{2}$	7
24	27.14	45.24	$5\frac{3}{4}$	$1\frac{3}{16} \times 22\frac{7}{8}$	$2 \times 22\frac{7}{8}$	12	15	$3\frac{1}{2}$	6	$7\frac{1}{2}$
26	31.85	53.09	$6\frac{1}{8}$	$1\frac{5}{16} \times 24\frac{3}{4}$	$2\frac{1}{8} \times 24\frac{3}{4}$	13	$16\frac{1}{2}$	$3\frac{3}{4}$	$6\frac{1}{2}$	$8\frac{1}{2}$
28	36.95	61.58	$6\frac{1}{2}$	$1\frac{7}{16} \times 25\frac{1}{2}$	$2\frac{1}{2} \times 25\frac{1}{2}$	14	$17\frac{1}{2}$	4	7	9
30	42.41	70.69	7	$1\frac{9}{16} \times 27\frac{1}{2}$	$2\frac{5}{8} \times 27\frac{1}{2}$	15	19	$4\frac{1}{2}$	$7\frac{1}{2}$	$9\frac{1}{2}$
32	48.26	80.43	$7\frac{1}{2}$	$1\frac{5}{8} \times 29\frac{1}{4}$	$2\frac{3}{4} \times 29\frac{1}{4}$	16	20	5	8	$10\frac{1}{2}$
34	54.48	90.79	8	$1\frac{3}{4} \times 31\frac{1}{4}$	$2\frac{7}{8} \times 31\frac{1}{4}$	17	$21\frac{1}{2}$	$5\frac{1}{2}$	$8\frac{1}{2}$	11
36	61.07	101.79	$8\frac{1}{2}$	$1\frac{7}{8} \times 33$	$3\frac{1}{8} \times 33$	18	$22\frac{1}{2}$	6	9	$11\frac{1}{2}$

TABLE V

PROPORTIONS OF CORLISS ENGINES

Diameter of Cylinder Inches	Thickness of Cylinder Inches	Thickness of Chests Inches	Thickness of Valve Chamber Inches	Bearing of Valves Inches
10	$\frac{3}{4}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{7}{8}$
12	$\frac{13}{16}$	$\frac{5}{8}$	$\frac{3}{4}$	$1\frac{1}{16}$
14	$\frac{7}{8}$	$\frac{11}{16}$	$\frac{13}{16}$	$1\frac{1}{8}$
16	$\frac{15}{16}$	$\frac{3}{4}$	$\frac{7}{8}$	$1\frac{1}{4}$
18	1	$\frac{13}{16}$	1	$1\frac{3}{8}$
20	$1\frac{1}{16}$	$\frac{7}{8}$	$1\frac{1}{16}$	$1\frac{1}{2}$
22	$1\frac{1}{8}$	$\frac{15}{16}$	$1\frac{1}{8}$	$1\frac{5}{8}$
24	$1\frac{3}{16}$	$\frac{15}{16}$	$1\frac{1}{8}$	$1\frac{3}{4}$
26	$1\frac{1}{4}$	1	$1\frac{3}{16}$	$1\frac{7}{8}$
28	$1\frac{1}{4}$	1	$1\frac{1}{4}$	2
30	$1\frac{5}{16}$	$1\frac{1}{16}$	$1\frac{5}{16}$	$2\frac{1}{8}$
32	$1\frac{3}{8}$	$1\frac{1}{8}$	$1\frac{3}{8}$	$2\frac{1}{4}$
34	$1\frac{3}{8}$	$1\frac{1}{8}$	$1\frac{3}{8}$	$2\frac{3}{8}$
36	$1\frac{3}{8}$	$1\frac{1}{8}$	$1\frac{3}{8}$	$2\frac{1}{2}$

50. An inspection of Tables II to V, inclusive, shows that the following rules are used in designing the line of engines considered:

Let D = diameter of cylinder. Then,

diameter of shaft = $.34\,D + 2\frac{1}{2}$ inches, nearly

A more exact but less simple formula, where L is the stroke, is

diameter of shaft = $.26\sqrt[3]{DL} + 2\frac{1}{2}$ inches

Diameter of crankpin = $\frac{D}{4}$.

Length of crankpin = $\frac{D}{4}$.

Length of crosshead pin = $\frac{D}{4}$.

Diameter of crosshead pin = diameter of crankpin × .65.

Diameter of steel valve stem = $.19\sqrt[3]{D^2}$.

Depth of piston

$= \frac{D}{4} + 1\frac{1}{2}$ inches for D less than 24 inches;

$= \frac{D}{8} + 4\frac{1}{2}$ inches for D greater than 24 inches.

Diameter of piston rod $= .16\,D$.

Width of crank-disk $= \frac{D}{4}$.

Area of steam port $= .06 \times$ area of cylinder.

Area of exhaust port $= .1 \times$ area of cylinder.

Diameter of valve $= \frac{3\,D}{16} + 1\frac{1}{4}$ inches.

Width of steam chest $= \frac{D}{2}$.

Width of exhaust chest $= .63\,D$.

Depth of steam and exhaust chests

$= \frac{D}{8} + \frac{1}{2}$ inch for D less than 28 inches;

$= \frac{D}{4} - 3$ inches for D greater than 28 inches.

Diameter of steam pipe $= \frac{D}{4}$.

Diameter of exhaust pipe $= .31\,D$.

Thickness of cylinder $= .028\,D + \frac{1}{2}$ inch.

Thickness of chests $=$ thickness of cylinder $\times .8$.

Thickness of valve chamber

$=$ thickness of chest $+ \frac{1}{8}$ inch for $D = 10$ to 16 inches;

$=$ thickness of chest $+ \frac{3}{16}$ inch for $D = 18$ to 26 inches;

$=$ thickness of chest $+ \frac{1}{4}$ inch for $D = 28$ to 36 inches.

Bearing of valve $=$ diameter of valve $\times .3$.

TYPES OF STEAM BOILERS

STATIONARY AND MARINE BOILERS

CLASSIFICATION OF BOILERS

DEFINITIONS

1. A **steam boiler** is a closed vessel in which steam is generated for power or heating purposes. The boiler when in use is but partially filled with water, the space within it being thus divided into two parts.

The **water-line** is the level of the surface of the water in the boiler.

The **steam space** is the space in the boiler above the water-line.

The **heating surface** of a boiler is the part of its surface that is exposed to the fire and to the hot gases from the fire as they pass from the furnace to the chimney.

The **fittings** of a boiler consist of such attachments as gauges for showing the steam pressure and the amount of water in the boiler, the safety valve, the steam stop-valve, etc.

2. Steam boilers may be classified according to their form, construction, and use. Thus, according to their form, boilers are *horizontal* and *vertical;* according to their construction, they are *shell*, *flue*, *sectional*, *fire-tube*, and *water-tube* boilers; according to the different conditions under which they are used, they are designated as *stationary*, *locomotive*, and *marine* boilers.

A **shell**, or **cylindrical, boiler** is one consisting of a plain cylinder closed at both ends.

A **sectional boiler** is one made up of a number of small sections.

A **flue boiler** has one or more large flues or pipes, 12 inches or more in diameter, surrounded by water and so arranged that the hot gases must pass through the flues.

A **fire-tube boiler** resembles a flue boiler in possessing flues, but differs from it in that these flues are numerous and small, being 6 inches or less in diameter. The hot gases pass through these smaller flues or tubes just as they pass through the larger flues of a flue boiler.

A **water-tube boiler** consists of a number of tubes connected to drums and so arranged that water circulates within them while the heating is done by the hot gases surrounding them.

The main features of different types of boilers are frequently combined, giving rise to a large number of special forms.

STATIONARY BOILERS

HORIZONTAL SHELL, FLUE, AND TUBULAR TYPES

3. A plain cylindrical boiler, Figs. 1 and 2, consists essentially of a long cylinder or **shell** made of iron or steel plates riveted together, the girth seams having a single row of rivets and the longitudinal seams a double row of rivets. The shells of boilers of this type are usually from 30 inches to 40 inches in diameter, and from 20 feet to 40 feet in length, although in some cases the length has been made as great as 70 feet. The **heads**, or ends, of the cylinder are either hemispherical or flat. The former are generally used, as they are stronger than flat heads and require no bracing. They are sometimes made of cast iron, but wrought iron or steel plate is preferable. The manner of suspending the shell is shown in Figs. 1 and 2.

The boiler is supported and enclosed by side walls of brick, known as the boiler setting. The channel beams *i*, *i* are laid

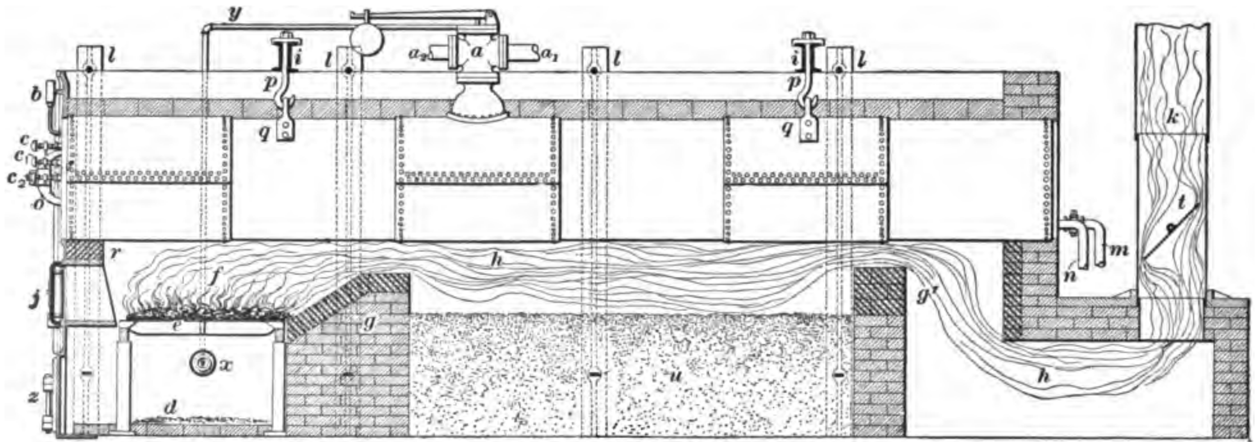
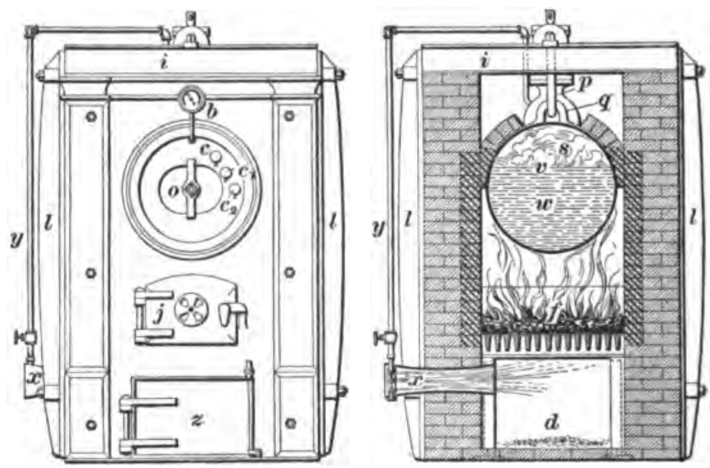


Fig. 1

across these brick side walls, and the boiler is suspended from the beams by means of the hooks *p*, *p* and eyes *q*, *q*, the latter being riveted to the shell.

The side walls are supported and prevented from buckling by the **binders** or **buckstaves** *l*, *l*, bolted together at the top and the bottom. The buckstaves are cast-iron bars of **T** section. The eyes *q*, *q* are placed about one-fourth of the length of the shell from each end. This method of suspending the shell allows it to expand and contract freely when heated or cooled.

The rear wall is built around the rear end of the shell, as



(*a*) (*b*)

FIG. 2

shown in Fig. 1, and continued back to form the chamber *h*, into which opens the chimney or stack *k*. The **boiler front**, shown in Fig. 2 (*a*), is of cast iron. Fig. 1 shows a section of the front. The front end of the shell is partly surrounded by the firebrick *r*, but the weight of the shell comes on the hooks *p*, *p*, the rear wall and firebrick *r* simply keeping the shell in position.

The furnace *f* is placed under the front end of the boiler shell. The fuel is thrown in through the furnace door *j* and burns on the grate *e*, the ashes falling through the grate into the ash-pit *d*. To insure a supply of air sufficient for the complete combustion of the fuel, the furnace is sometimes provided with a blower *x*, consisting of a cylinder leading into the ash-pit *d*, into which is led a jet of steam through the pipe *y*. The steam rushes into the ash-pit with great velocity and carries a quantity of air with it. The pressure of the air in the ash-pit is thus increased, more air is forced through the fire, and the combustion of the fuel is more rapid and complete.

Behind the furnace is built the brick **bridge wall** *g*, which serves to keep the hot gases in close contact with the under side of the boiler shell. As boilers of this type are generally quite long, a second bridge wall g' is usually added. The gases arising from the combustion of the fuel flow over the bridge walls g, g' into the chamber *h*, and escape through the chimney *k*. The flow of the gases is regulated by the damper *t* placed within the chimney. The space *u* between the bridge walls is filled with ashes or some other good non-conductor of heat. The door *z* in the boiler front gives access to the ash-pit for the removal of the ashes. The tops of the bridge walls, the inner surfaces of the side walls and rear wall, and, in general, all portions of the brickwork exposed to the direct action of the hot gases, are made of fire-brick as shown by the dark section lining in Figs. 1 and 2 (*b*), since it is able to withstand a very high temperature.

The brickwork covers the upper portion of the boiler shell in such a manner as to prevent the hot gases from coming in contact with the shell above the water-line *v*. It is a general rule in boiler construction and setting that *under no circumstances should the fire-line be carried above the water-line.*

The top of the shell is covered by brickwork or some other non-conducting material to prevent radiation of heat. Water is forced into the boiler through the feedpipe *n* that leads from a pump or an injector. When in operation, the water stands at about the level *v*, the space *s* above being occupied by the steam. The **safety valve** is shown at *a*; its office is to prevent the steam pressure from rising above the desired point. The pipe a_1 is the main steam pipe leading to the engine; the pipe a_1 provides for the escape of the waste steam when the safety valve blows off. The **steam gauge** *b* indicates the pressure of the steam in the boiler: this gauge is attached to a pipe that passes through the front head into the steam space.

The **gauge-cocks** c, c_1, and c_2, placed in the front head of the shell, are used to determine the water-level. If any one of the cocks is opened and water escapes, it is evident that the water-line is above that cock, while if steam escapes, the level must be below it.

The **manhole** *o* is a hole in the front head through which a man may enter and inspect or clean the boiler; it is closed by a plate and yoke.

To permit the boiler to be emptied, it is provided with a **blow-off pipe** *m*, through which the water and sediment may be discharged.

Plain cylindrical boilers are much used in mining districts, where fuel is very cheap, but on account of their small heating surface they are very uneconomical, and consequently are not generally used where fuel is expensive. The advantages of this type of boiler are: cheapness of construction, strength, durability, and ease of access for cleaning and repairs.

4. The **flue boiler** differs from the plain cylindrical boiler in having one or more large flues running lengthwise through the shell, below the water-line. Such a boiler is shown in elevation and section in Figs. 3 and 4.

The ends of the flues *a, a* are fixed in the front and rear heads of the shell. The front end of the shell is prolonged beyond the head, forming the **smokebox** *b*, into which opens the smokestack *c*. The front of the smokebox is provided with a door *e*. The boiler shell is also provided with the **dome** *d*, which forms a chamber where steam may collect and free itself from its entrained water before passing to the engine. The manner of supporting the shell and the construction of the furnace and bridge walls are the same as was described for the plain cylindrical type of boiler. The hot

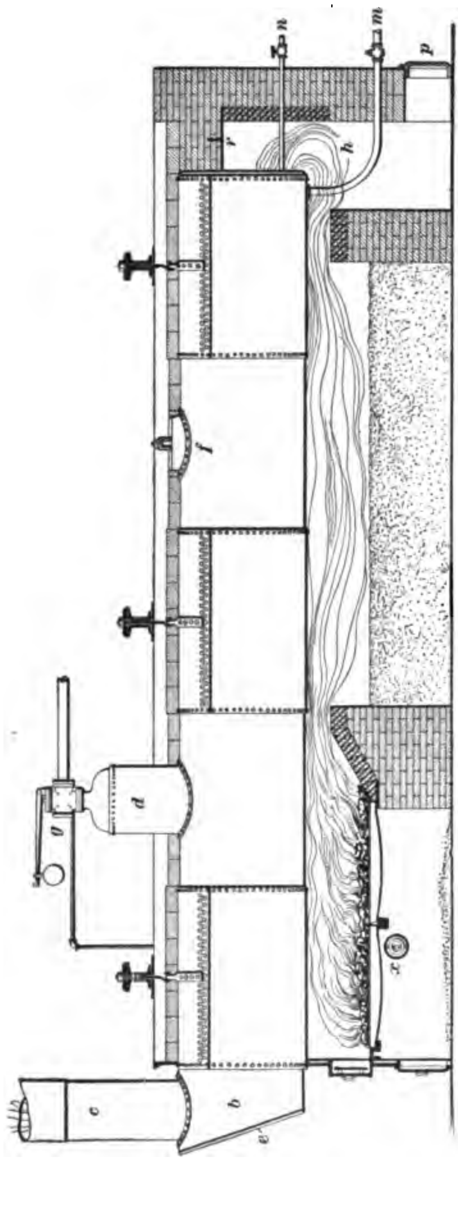
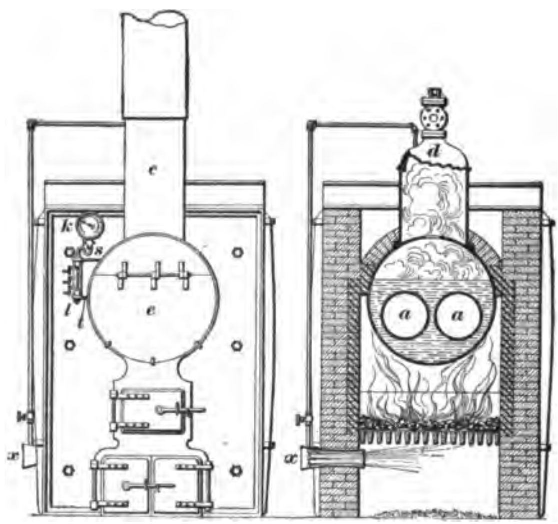


FIG. 8

gases, however, pass over the bridge walls to the chamber *h*, and then back through the flues *a*, *a* into the smokebox *b*, and out of the stack *c*. It is plain, therefore, that the heating surface is greater than that of the plain cylindrical boiler by the cylindrical surfaces of the flues *a*, *a*.

The boiler has a cast-iron front, to which the furnace door and ash-pit doors are attached. The safety valve *g* is attached to the top of the dome; from the safety valve are



(a) (b)

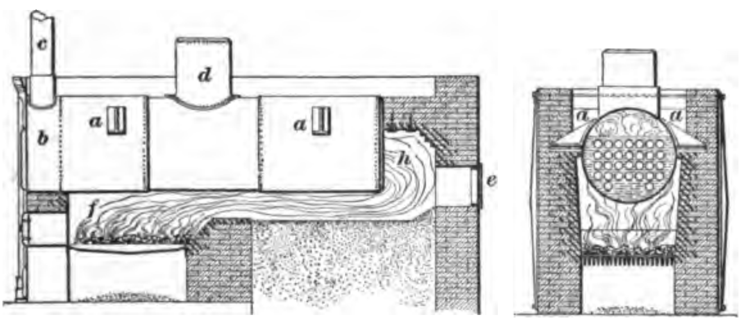
Fig. 4

led the two steam pipes, one to the engine, the other to carry the escaping steam outside the building.

The steam gauge *k* and gauge-cocks are placed on a column *l* that communicates with the interior of the shell through the pipes *s* and *t*, the former entering the steam space and the latter the water space. The manhole *f* is placed on top of the shell instead of in the head. The feedpipe is shown at *n*, and the blow-off pipe at *m*, both passing through the rear wall. Access is given to the rear end of the shell and to the pipes *m* and *n* through the door *p*. This form of boiler may be provided with a blower, as shown at *x*.

The setting is built and supported in about the same manner as that shown in Fig. 1. The cast-iron **flue plate** *r* rests on the side and rear walls and supports the brickwork above it.

5. Horizontal Return-Tubular Boiler.—By extending the principle of the flue boiler, that is, by making the flues more numerous and smaller, the horizontal return-tubular boiler has been developed. In this boiler, by far the greatest amount of the total heating surface is provided by the small flues, called **tubes,** that are traversed by the hot gases from the furnace. The space occupied is moderate in comparison with a flue boiler of equal steam-gener-



(a) (b)

FIG. 5

ating capacity, although greater than that required for water-tube boilers.

A side view of a horizontal return-tubular boiler is shown in Fig. 5 (*a*); a cross-section through the tubes is shown in Fig. 5 (*b*). The tubes extend through the whole length of the shell, the ends being expanded into holes in the heads of the boiler. The front end of the shell projects beyond the head, forming the smokebox *b*, into which opens the stack *c*.

The shell is supported on the side walls by the brackets *a*, *a* which are riveted to the shell. The boiler is usually provided with a dome *d*, though this is sometimes omitted. The walls are built and supported by buckstaves in practically the same manner as those previously described. Since this type of boiler is generally short, one bridge only is used. Firebrick is used for all parts of the wall exposed to the fire or heated gases. The fittings are not shown in the figure. The safety valve would be placed on top of the dome, and the pressure gauge and gauge-cocks would be placed on the boiler front. The manhole is either in one of the heads or on top of the shell. The feedpipe may enter the front head, the rear head, or at the rear end of the bottom of the shell; the blow-off pipe is placed at the bottom of the shell at the rear end. Access is given to the rear end of the boiler through the door *e*.

As usual, the furnace *f* is placed under the front end of the boiler. The gases pass over the bridge wall, along under the boiler into the chamber *h*, then back through the tubes to the smokebox *b*, and out of the stack *c*.

6. Cornish Boiler.—In the three forms of boilers so far considered, the furnace is placed outside the shell of the boiler; such boilers are said to be **externally fired.** On the invention of the single-flue boiler, the idea was conceived of placing the fire in the flue, and the result is the so-called **Cornish boiler.**

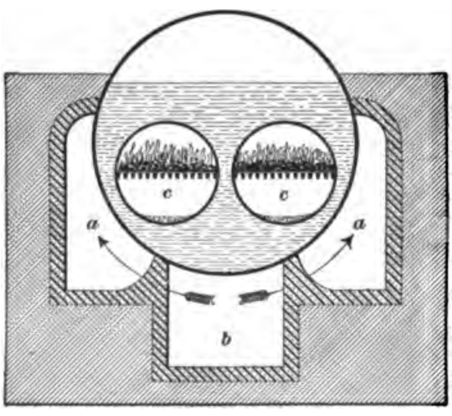


FIG. 6

7. Lancashire Boiler.—The **Lancashire boiler,** as shown in Fig. 6, is a modification of the Cornish type. In order to give a large grate area and a large heating surface for the same diameter of shell, two large furnace flues are substituted for the one flue of the Cornish type. The boiler is set in masonry in such a manner as to form the passages *a*, *a*, and *b*. The grates are supported in the flues *c*, *c*. The heated gases pass from the furnaces to the rear end through the flues *c*, return beneath the boiler through the flue *b*, and finally return to the rear through the side flues *a*, *a*, and thence out of the chimney. This path of the gases constitutes the **split draft.**

8. The large furnace flues of **internally fired** boilers, of which the Cornish and Lancashire are examples, are subjected to an external collapsing pressure, equal to the pressure of the steam. The greater the diameter of the flue, the more likely it is to collapse; consequently, the Lancashire possesses an advantage over the Cornish type in this respect, since each of its two flues is necessarily of smaller diameter than the single flue of the Cornish boiler. Various measures are taken to strengthen the furnace flues of internally fired boilers. They are sometimes made corrugated, and sometimes they have channel irons riveted around them. A very common method, however, is to stiffen them by transverse conical water legs, as shown in Fig. 7.

9. Galloway Boiler.—The **Galloway boiler** is a sort of combination of the Cornish and Lancashire types. It has two internal furnace flues fitted with grates, ash-pit, etc. in the usual manner. Instead of extending through the whole length of shell, the two flues unite just behind the bridge into one large flue that extends from this junction to the rear head of the shell. This large flue is strengthened by a large number of water legs of the form shown in Fig. 7. The setting of the Galloway boiler is similar to that shown in Fig. 6. The draft is split as described in connection with the Lancashire boiler.

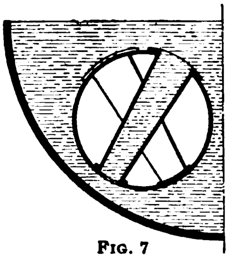


FIG. 7

10. Clyde Boiler.—A stationary boiler combining the features of the Lancashire and multitubular types is shown in Fig. 8. It is known as the **Clyde boiler** and consists of a large cylindrical shell that is short in comparison with its diameter. Two large furnace flues *A* (only one of which is shown in the figure) extend side by side lengthwise through the shell, the ends of the flues being riveted to the heads of the shell. Above and parallel with these flues, and below the water-line, a series of tubes *B* extend through the shell from end to end. The front ends of these tubes open into a smokebox *F*, which is connected with the chimney or stack, and the rear ends of the tubes connect with the combustion chamber *E*.

The boiler is enclosed in brickwork, which, however, does not form the support, but is built to form the chamber *E* and to prevent radiation. The shell is supported on the beams *O*, *O*, *O*.

The furnace flues are corrugated to render them stronger against a collapsing pressure. The flat heads are restrained

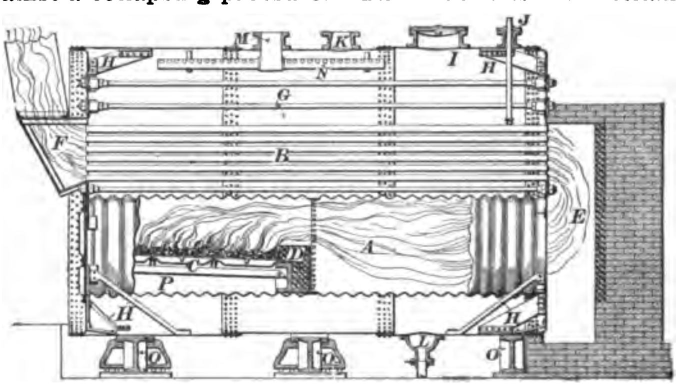


FIG. 8

from bulging by the heavy stayrods *G* and the diagonal braces *H*, *H*.

The furnaces are placed within the large flues, and, as usual, consist of the grate *C*, ash-pit *P*, and bridge wall *D*. The gases arising from the combustion flow to the rear through the flues *A* into the chamber *E* and back to the front through the tubes *B*, thence into the smokebox *F* and out of the chimney. This type of boiler has a very large amount of heating surface in proportion to its grate area.

Water is fed into the boiler through the feedpipe *J*; the various fittings are not shown in the figure. The pressure gauge and gauge-cocks are usually placed on the boiler front; the safety valve is bolted on to the nozzle *K*; the blow-off is at L. The main steam pipe is bolted on at M. The **dry pipe** N is a device for freeing the steam from any unevaporated water which may be mixed with it. It is often used in place of a dome on this form of boiler, and also on the other types of internally fired boilers just described. The manhole is shown at I.

11. Locomotive or Firebox Boiler.—Next to the multitubular type, the **locomotive** or **firebox boiler** is probably more used than any other type. For railway service it is used exclusively and is also largely used as a stationary boiler. Small portable boilers used for agricultural purposes are usually of the firebox type. The general construction of this type of boiler is shown in Fig. 9. The

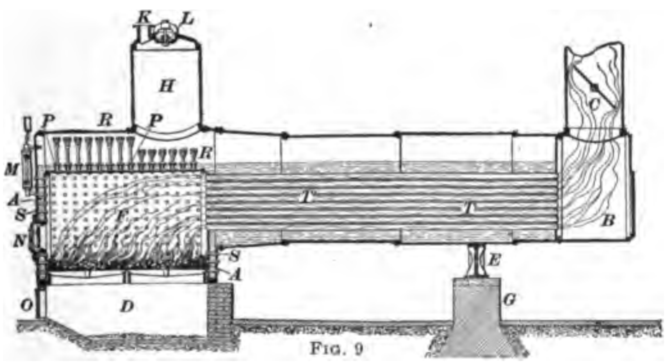


FIG. 9

shell is composed of two differently shaped parts riveted together. The end B is known as the front end, since, when used on a locomotive, it stands toward the front. The front part of the shell is cylindrical; the rear part is usually of a rectangular cross-section with vertical sides, or of a trapezoidal section with inclined sides; in either case, the top is semicylindrical. The furnace F is a box of the same shape as the rear end of the shell in which it is placed. A space, called a *water leg*, is left between the sides and ends of the furnace and the shell; this space is filled with water as shown at A, A. A series of tubes T, T extend from the front sheet of the furnace or firebox to the front head of the shell.

The shell is prolonged beyond the front head, forming a smokebox *B*, which opens into the stack *C*.

In this figure, the spaces *A*, *A* extend down only as far as the grate, the ash-pit *D* being formed in the brick setting. In many boilers of this type, the water legs extend down to the bottom of the ash-pit, and sometimes there is a water space below the ash-pit; that is, the furnace and ash-pit are entirely surrounded by water.

The boiler is supported by a cast-iron cradle *E* that rests on the masonry foundation *G*, and by a brick wall, which also forms the ash-pit. The boiler is usually provided with a dome *H*, from which is led the main steam pipe, bolted on at *K*. In the figure, the dome is provided with a manhole *L*. The feedwater may be introduced at any convenient point in the shell. The pressure gauge, water glass, and gauge-cocks are attached to the column *M*, which is in communication with the interior of the shell. The furnace and ash-pit doors are shown at *N* and *O*, respectively. The safety valve is usually attached to the dome.

Since the flat sides of the furnace and shell are liable to bulge on account of the pressure, they must be braced or stayed; this is accomplished by the staybolts *S*, *S*. The flat top of the firebox is strengthened by a series of parallel girders *P*, *P*. As an additional security, the girders are sometimes attached to the shell by the sling stays *R*, *R*.

The gases of combustion pass directly from the furnace through the tubes *T*, *T*, to the smokebox *B*, and out of the stack *C*. In railway locomotives, a strong draft is obtained by allowing the exhaust steam to discharge through the smokestack.

The locomotive type of boiler is *self-contained*, that is, it requires no brickwork for flues or for setting.

VERTICAL TUBULAR BOILERS

12. The **vertical** type of fire-tube boiler may be considered as a modification of a locomotive type placed on end, and, in common with that type, is self-contained.

A common form of vertical boiler is shown in Fig. 10. It consists of a vertical cylindrical shell, in the lower end of which is the firebox *F*. The lower rim of the firebox and the lower end of the shell are separated by a wrought-iron ring *k* to which both are riveted, the rivets going through both plates and ring. The shell and firebox are also stayed together by the staybolts *a*, *a*. The space between the two is filled with water, so that the firebox is nearly surrounded by it. The boiler shell, and likewise the grate *E*, rests on a cast-iron base *D* that forms the ash-pit. A series of vertical tubes *t*, *t* extend from the top sheet of the firebox to the upper head of the shell. The tubes serve as stays, strengthening the flat surfaces that they connect. The upper ends of the tubes open directly into the chimney, or smokestack, *K*. The gases from the furnace thus pass directly through the tubes and out of the stack. The safety valve is shown at *H*, with the main steam pipe *G* leading from it. The pressure gauge *P* and gauge-cocks *c*, *c*, *c* are attached to a column *L*, which communicates in the usual manner with the interior of the shell. The construction of this type of boilers does not generally permit the use of manholes, but handholes *M*, *M* are placed in convenient positions for cleaning out mud and sediment.

13. When the tubes extend through the upper head of the boiler, as shown in Fig. 10, their upper ends pass through the steam space *S* above the water-line *V*, *V*. This is looked on as a bad feature, since the tubes are liable to become overheated and to collapse when the boiler is forced.

In the form of vertical boiler shown in Fig. 11, this danger is avoided. A chamber, or smokebox, *I* extends down from the upper head of the shell, so that its bottom plate is always below the water-line. The upper ends of the tubes *t*, *t* are expanded into the lower plate of this chamber, and therefore the tubes are always surrounded by water from end to end. A vertical boiler constructed in this manner is said to have a *submerged head*. Aside from the submerged head, the construction of the boiler is similar to that shown in Fig. 10.

Vertical boilers are generally wasteful of fuel, and are perhaps more likely to explode than any other type. They are, however, self-contained, require but little floor space, and are easy to construct and repair. For these reasons, they are very popular for small installations.

14. Boilers having the construction shown in Figs. 10

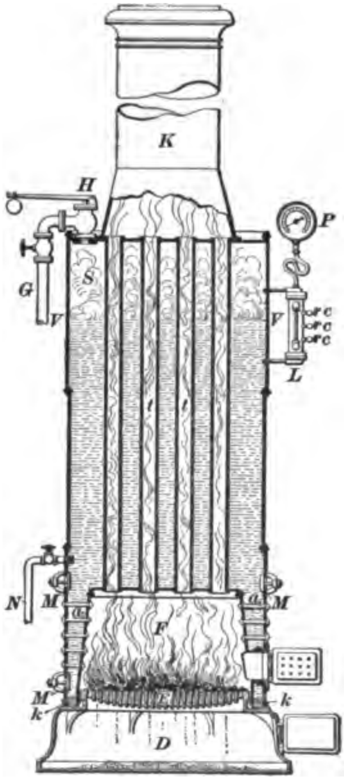


Fig. 10

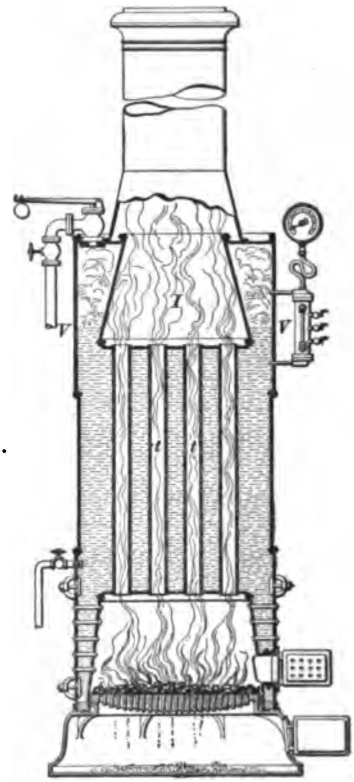


Fig. 11

and 11 are liable to throw sparks from the chimney, especially when the fire is forced. This makes their use dangerous in many localities.

HORIZONTAL WATER-TUBE BOILERS

15. Advantages of Water-Tube Boilers.—The boilers previously described have been of the types in which the water surrounds the tube or tubes, the flame and hot gases being inside the tube. In the **water-tube boiler,** this condition is reversed; the water is inside the tubes, which are surrounded by the fire and hot gases. Water-tube boilers are commonly known as *safety boilers*, because an accident to any one tube or fitting does not necessarily involve the destruction of the whole boiler. They are extensively used for land service, and are being introduced into marine service. The demand for very high steam pressures has led to the development of the water-tube boiler, which is not a recent invention, as the first patents for this type of boiler were issued in 1788.

It is maintained that the heating surface in water-tube boilers is much more effective than an equivalent area of surface in the ordinary tubular boilers. In water-tube boilers, the direction of the circulation is well defined and there are no interfering currents. The circulation is rapid and over the entire boiler, keeping it at a nearly constant temperature and tending to deposit all the sediment at the lowest point. The water is divided into small bodies, the boilers steam quickly, and are sensitive to slight changes of pressure or conditions of the fire. The arrangement of a water-tube boiler is such as to form a flexible construction, any member being free to expand without unduly expanding any other member. This very important feature tends to prolong the life of the boiler.

There is considerable difference in the amount of soot collected in a fire-tube and on a water tube. Soot accumulates within a fire-tube, and it may become filled, while the water tube holds the soot only on the top surface. Water-tube boilers are of sectional construction, and hence may be transported and erected more readily than other types.

16. Babcock & Wilcox Boiler.—Water-tube boilers may be divided into straight-tube and bent-tube boilers.

One of the best known straight-tube water-tube boilers is the **Babcock & Wilcox boiler,** illustrated in Fig. 12. It consists essentially of a main horizontal drum *b* and of a series of inclined tubes *t*, *t*. Only a single vertical row of tubes is shown in the figure, but there are usually seven or eight of these vertical rows to each horizontal drum. The front ends of the tubes of a vertical row are all expanded into a hollow iron or steel fitting *h*, called a **header.** The rear ends are expanded into a similar header, and the front and rear headers are placed in communication with the drum

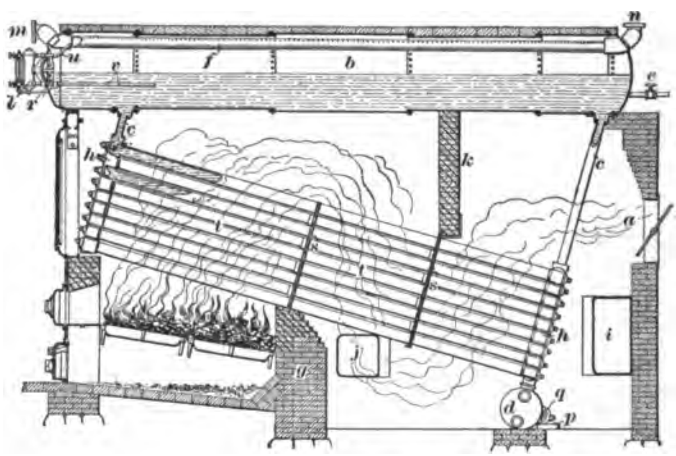


Fig. 12

by tubes, or **risers,** *c*, *c*. In the header, in front of each tube, a handhole is placed for the purpose of cleaning, inspecting, or removing the tube.

The method of supporting the boiler is not shown in the figure. The usual method is to hang the boiler from wrought-iron girders resting on vertical iron columns. The brickwork setting is not depended on as a means of support. This make of boiler, in common with most others of the water-tube type, requires a brickwork setting to confine the furnace gases.

The furnace is of the usual form and is placed under the front end of the nest of tubes. The bridge wall *g* is built up to the bottom row of tubes; another firebrick wall *k* is built between the top row of tubes and the drum. These walls and the baffle plates *s*, *s* force the hot furnace gases to follow a zigzag path back and forth between the tubes. The gases finally pass into the chimney flue through the opening *a* in the rear of the wall. The feedwater is introduced through the feedpipe *e*. The steam is collected in the dry pipe *f*, which terminates in the nozzles *m* and *n*, to one of which is attached the main steam pipe, and to the other the safety valve. The pressure gauge, cocks, etc. are attached to the column *l*, which communicates with the interior of the shell by the small pipes *u* and *v*, the former of which extends into the dry pipe, the latter into the water. At the bottom of the rear row of headers is placed the mud-drum *d*. Since this drum is the lowest point of the water space, most of the sediment naturally collects there. This sediment may be blown out from time to time through the blow-off pipe *p*. The drum *d* is provided with a handhole *q*, and a manhole *r* is placed in the front head of the drum *b*. The heads of the drums are dished so as to form segments of a sphere, and therefore do not require bracing. Access may be had to the space within the walls through the doors *i* and *j*.

The circulation of water takes place as follows: The feedwater is introduced into the rear of the steam drum; the furnace being under the higher end of the tubes, the water in that end expands on being heated, and is also partly changed to steam; hence, a column of mingled water and steam rises through the front headers to the front end of the drum *b*, where the steam escapes from the surface of the water. In the meantime, the feedwater fed into the rear of the drum descends to the rear headers through the long tubes *c* to take the place of the water that has risen in front. Thus, there is a continuous circulation in one direction, sweeping the steam to the surface as fast as it is formed and supplying its place with cooler water.

17. Heine Boiler.—A boiler differing in many respects from that shown in Fig. 12 is the **Heine boiler**, illustrated in Fig. 13. It consists of a large main drum *a*, which is above and parallel with the nest of tubes *b*, *b*. Both drum and tubes are inclined at an angle with the horizontal that brings the water level to about one-third the height of the drum in front and to about two-thirds the height in the rear. The ends of the tubes are expanded into the large wrought-iron water legs *c*, *c*. These legs are flanged and riveted to the shell, which is cut out for about one-fourth of its circumference to receive them, the opening being from 60 to 90 per cent. of

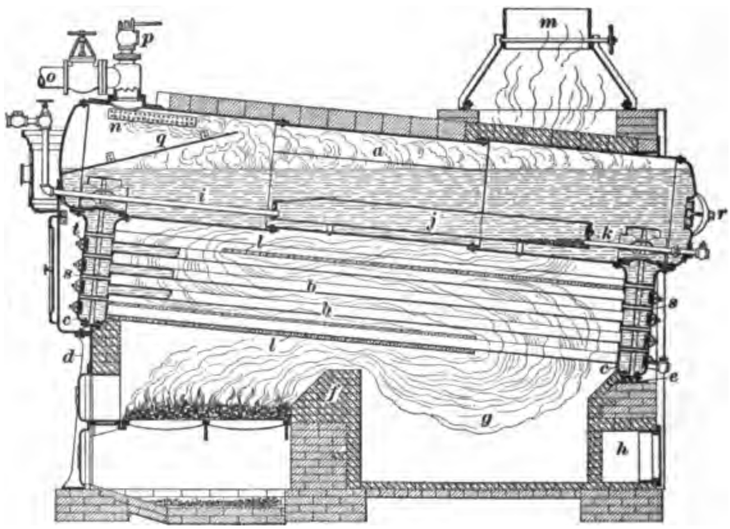


FIG. 13

the cross-sectional area of the tubes. The drum heads form segments of a sphere, and therefore do not need bracing. The water legs form the natural support of the boiler, the front water leg being placed on a pair of cast-iron columns *d* that form part of the boiler front, while the rear water legs rest on rollers, shown at *e*, which can move on a cast-iron plate bedded in the rear wall. These rollers allow the boiler to expand freely when heated. The boiler is enclosed by a brickwork setting in the usual manner. The bridge wall *f*, made largely of firebrick, is hollow, and has openings in the rear to allow air to pass into the chamber *g* and mix with the furnace gases. In the rear wall is the arched opening *h*, which is closed by a door and further protected by a thin wall of firebrick. When it is necessary to enter the chamber *g*, the wall at *h* may be removed and afterwards replaced. The feedwater is brought in through the feedpipe *i*, which passes through the front head. As the water enters, it flows into the mud-drum *j*, which is suspended in the main drum below the water-line and is thus completely submerged in the hottest water in the boiler. This high temperature is useful in precipitating the impurities contained in the feedwater, which settle in the mud-drum *j*, and may then be blown out through the blow-off pipe *k*. The water passes back out of the open end of the mud-drum and circulates in the same direction as in the boiler shown in Fig. 12. Layers of firebrick *l*, *l* are laid at intervals along the rows of tubes, and act as baffle plates, forcing the furnace gases to pass back and forth over the tubes. The gases finally escape through the chimney *m* placed above the rear end of the boiler. To protect the steam space of the drum from the action of the hot gases, the drum in the vicinity of the chimney is protected by firebrick, as shown in the figure. The steam is collected and freed from water by the perforated dry pipe *n*. The main steam pipe with its stop-valve is shown at *o*, and the safety valve at *p*. In order to prevent a combined spray of mixed water and steam from spurting up from the front header and entering the dry pipe, a deflecting plate *q* is placed in the front end of the drum. A manhole *r* is placed in the rear head of the drum. The flat sides of the water legs are stayed together by the staybolts *s*, *s*, which are made hollow to give access to the outside of the tubes. In front of each tube, a handhole *t* is placed to give access to the interior of the tube.

Where a battery of several of these boilers is used, an additional steam drum is placed above and at right angles to the drums *a*.

18. Stirling Boiler.—A well-known type of bent-tube stationary boiler is the **Stirling water-tube boiler**, shown in Fig. 14. It consists of a lower drum *a* connected with three upper drums *b, b, b* by three sets of nearly vertical tubes. These upper drums are connected by the curved tubes *c, c, c*. The curved forms of the different sets of tubes allow the different parts of the boiler to expand and contract freely without strain. The boiler is enclosed, as shown, in a brickwork setting, which is provided with various holes *h, h*, so that the interior may be inspected or repaired. The

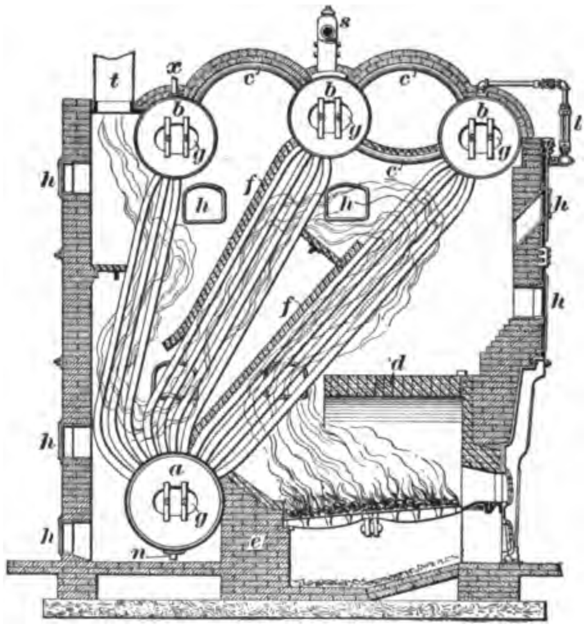


Fig. 14

boiler is suspended from a framework of wrought-iron girders, not shown in the figure. The bridge wall *e* is faced with firebrick, and is built in contact with the lower drum *a* and the front nest of tubes. An arch *d* is built above the furnace, and this, in connection with the bafflers *f, f*, directs the course of the heated gases, causing them to pass up and down between the tubes. The arch and bafflers are made of firebrick. The arch *d* becomes heated to a white heat, promoting combustion, and heating the incoming air when the furnace doors are opened, thus protecting the boiler from being chilled when the fires are being cleaned or stoked. The cold feedwater enters the rear upper drum through the pipe *x* and descends through the rear nest of tubes to the drum *a*, which acts as a mud-drum, and collects the sediment brought in by the water. A blow-off pipe *n* permits the removal of the sediment. From the drum *a*, the water passes upwards, through the two forward sets of tubes and is vaporized as it rises, the steam passing from the front drum to the middle drum through the upper set of curved tubes *c*, while the unvaporized water circulates between the front and middle drums through the lower set of curved tubes *c*, and thus the heated water does not again mingle with the comparatively cold water in the drum *a*. The steam collects in the upper drums *b*. The steam pipe and safety valve *s* are attached to the middle drum. The chimney *t* is located behind the rear upper drum. The water column *l*, with its fittings, is placed in communication with the front upper drum. All the drums are provided with large manholes *g*. The boiler is made with a cast-iron front.

VERTICAL WATER-TUBE BOILERS

19. Hazelton Boiler.—Another form of water-tube boiler, known as the **Hazelton boiler,** is shown in Fig. 15. It is sometimes called a **porcupine boiler,** because of the peculiar arrangement of the tubes. It consists of a central vertical cylinder, or shell *a*, with a large number of radial tubes *b* having their inner ends expanded into holes in the shell and their outer ends closed. The grate *c* surrounds the cylinder near the bottom. The inner ends of the grate bars rest on a ring *d* supported by brackets riveted to the shell; the outer ends rest on a plate on the brickwork enclosing the ash-pit. The boiler rests on a circular cast-iron base *e* placed on a masonry foundation. The boiler and furnace are enclosed in brickwork that supports the chimney. The brickwork is built up square to the height of the lower tubes and circular above that point. The furnace brickwork

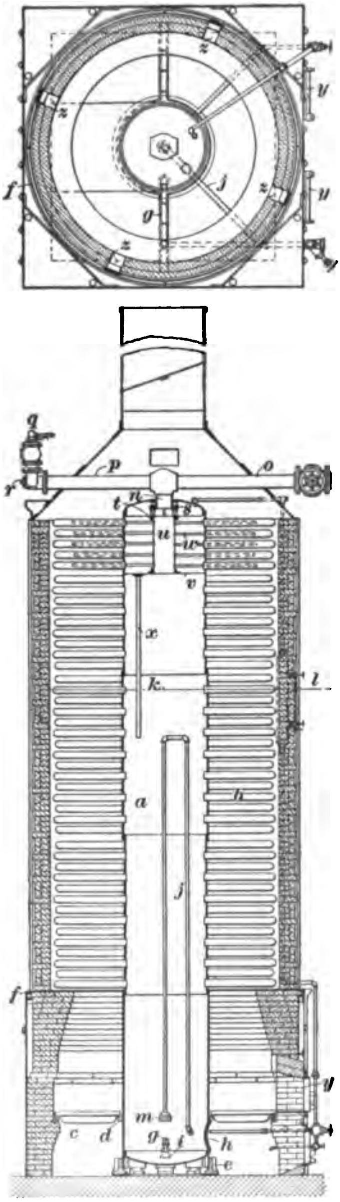


FIG. 15

is encased in sheets of steel riveted to angle irons at the corners and reenforced by angle and T bars riveted to them. An air space is usually provided between the brick lining and the casing to decrease the radiation.

The top of the furnace casing supports a circular plate *f*, on which is built the brick casing above the furnace. The circular brick casing is enclosed in sections of sheet steel bolted together. The firebrick lining of the furnace is built so as to slope inwards at the top and deflect the flame against the stand pipe of the boiler. The lower end of the stand pipe below the grates forms a settling chamber and mud-drum. It is fitted with a blow-off pipe *g* and a manhole *h* opposite one of the ash-pit doors. The blow-off pipe enters the mud-drum below the grate and terminates in a cone-shaped nozzle *i*, over the center of, and close to, the lower head. The feed-pipe *j* enters the shell below the grate and extends vertically nearly to the water-line *k l* in the boiler. It then passes downwards through the axis of the boiler and delivers the water through a spraying nozzle *m* at the level of the grate. The steam outlet is through a heavy nipple *n* screwed through the center of the top head of the steam drum. A **T** on the outer end of the nipple provides openings for the steam pipe *o* and a pipe *p* leading to the safety valve *q*. A handhole *r* is located on the end of the pipe below the safety valve, which is uncovered to afford ventilation to the interior of the boiler when it is necessary for a man to enter it. The nipple *n* terminates at its lower end in a flange *s*, to which is bolted a blank flange *t* at a distance of several inches. This blank flange closes the top of a short length of large pipe *u* suspended from it.

A diaphragm *v* is attached to the lower end of the pipe *u* and the shell of the boiler and closes the annular space between them. From the central pipe *u* a large number of small pipes *w* radiate horizontally and extend into the boiler tubes nearly to their outer ends. The steam flows from the central pipe through the small pipes into the boiler tubes, and thence backwards into the top of the steam drum, whence it passes out between the two flanges *s* and *t*. A drip pipe *x* is suspended from the diaphragm and extends a short distance below the water level in the boiler. Two firing doors *y* are located at one side of the furnace, and several doors are conveniently located in the brick setting, so that an examination can be made of the exterior of the boiler shell and tubes.

20. Morrin Climax Boiler.—The **Morrin Climax boiler**, shown in Fig. 16, is a water-tube boiler that somewhat resembles the porcupine boiler, and differs from it chiefly in that the stand pipe or main shell *a* is fitted with a large number of loop-like tubes *b*, *b*, instead of radial tubes. The ends of the tubes are expanded into the shell. The furnace is circular, as in the porcupine boiler, and in order to give free access to the fire, four furnace doors are provided. A deflector plate *d* is fitted to the shell a little above the water level, which tends to throw back any water carried up by the steam. The upper portion of the central shell is divided by a series of diaphragms *e*, *e* into a series of superheating chambers, through which the steam is compelled to circulate successively by the connecting loop-like tubes. The steam thus

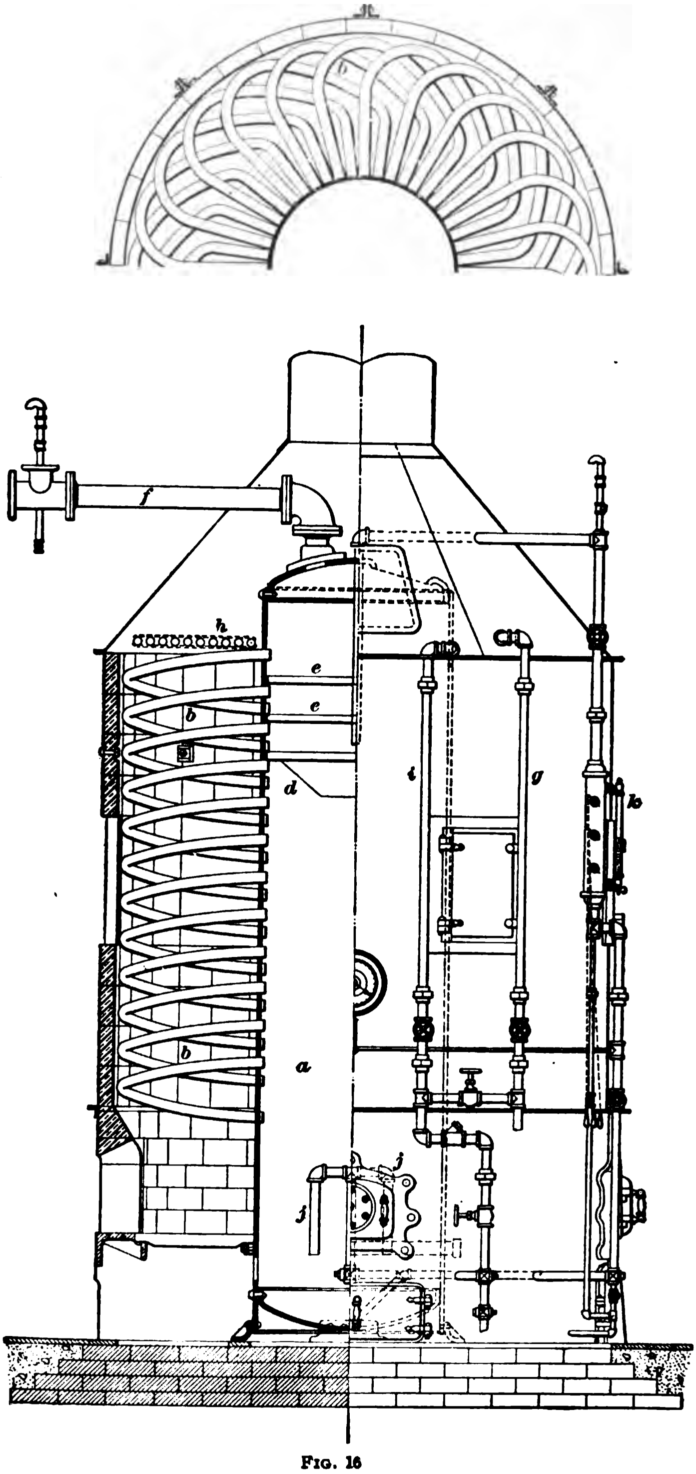


FIG. 16

becomes thoroughly dried and somewhat superheated before it enters the main steam pipe *f*. The feedwater passes through the pipe *g* into a spiral feed-coil *h* resting on top of the tubes, where it is heated to a high temperature. It leaves the coil through the pipe *i*, passes downwards, and enters the bottom of the shell through the internal feedpipes *j*, *j*. The water column *k* is connected to the top and bottom of the central shell. The safety valve is attached to a **T** placed in the main steam pipe close to the boiler.

21. Wickes Boiler.—Another form of vertical water-tube boiler, known as the **Wickes boiler,** is shown in Fig. 17. It consists of two drums *a* and *b* united by a number of long straight tubes *c* and encased in brickwork. The tubes are arranged in two banks separated by a vertical tile deflector *d*. The furnace *e* is located at one side of the boiler and the products of combustion pass upwards along the first section of tubes, over the top of the deflector *d*, down along the second section of tubes, and out through the opening *f* near the base of the setting to the chimney *g*. The top drum *a* is called the steam drum and the lower one *b* the mud-drum. Each drum has a dished head provided with a manhole. The feedwater enters the steam drum *a* near the bottom, through a pipe *h* at the side farthest from the furnace. The water-line in the steam drum is at a sufficient height to completely submerge the water tubes. The water passes down the rear bank of tubes, called *downcomers*, into the lower drum, and then up through the front bank of tubes, called *risers*, in front of the deflector *d*. A plate is placed above the risers in the steam drum to deflect the water to the rear and prevent particles of water from being carried through the steam outlet *i* in the center of the top of the drum. The boiler is supported by brackets *j* fastened to the mud-drum and resting on the brickwork. The boiler and the setting are free to expand independently of each other. Several doors *k* are placed in the setting so that the exterior of the boiler may be inspected or cleaned. Boilers of this type are also arranged horizontally.

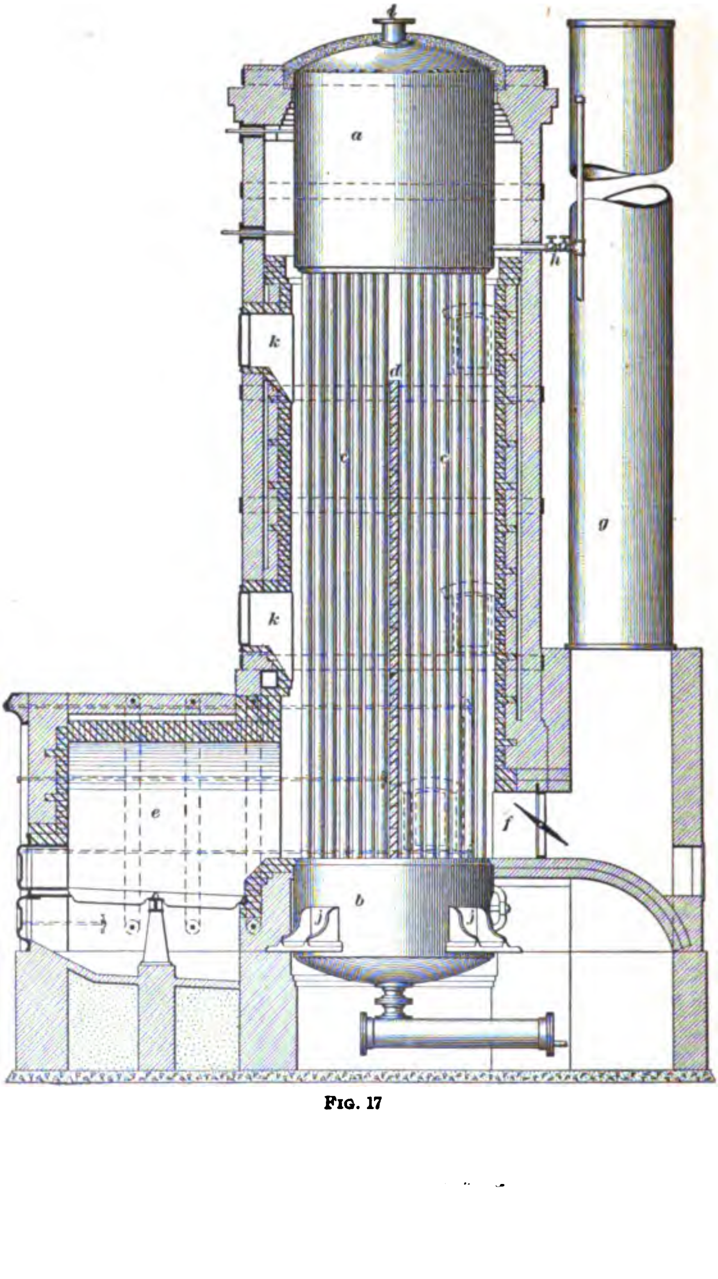


Fig. 17

MARINE BOILERS

SCOTCH MARINE BOILERS

22. Marine boilers present the greatest variety of form and construction. Each kind of service requires a boiler particularly suited to it. In large passenger and freight steamers, the Scotch, or drum, boiler is the favorite. High-speed torpedo boats and torpedo-boat destroyers are generally fitted with some form of water-tube boiler. Battle ships are usually fitted with drum boilers, but in some cases with water-tube boilers. A rectangular or box form of boiler is sometimes used for low pressures, seldom exceeding 30 pounds. For pressures of about 125 pounds, the cylindrical return-tubular boiler is sometimes used. The locomotive type of boiler is found on many small steamers, and is used for pressures up to 250 pounds.

The **Scotch boiler** is distinctly a marine boiler. It is generally fitted with two, three, or four internal furnaces. It is made from 10 to 20 feet in diameter, and from 7 to 12 feet long, and may be either single or double ended. Scotch boilers are combined in various ways, as regards the arrangement of tubes and combustion chambers, to secure lightness and efficiency. They require extremely careful handling to prevent internal stresses and strains that cause leaks.

23. Single-Ended Scotch Boiler.—A **single-ended Scotch marine boiler** with a cylindrical shell and flanged heads is shown in Fig. 18. It has four corrugated furnace flues *a*, *a* (of which but two are shown), opening into combustion chambers *b*. Usually, each flue has its own combustion chamber, but in some cases two or more flues open into a common chamber. A nest of fire-tubes *t*, *t* extends from the front plates of the combustion chambers to the front head of the shell. These flues place the combustion chambers *b* in communication with the large smoke chamber *e*, which, in turn, leads directly to the stack.

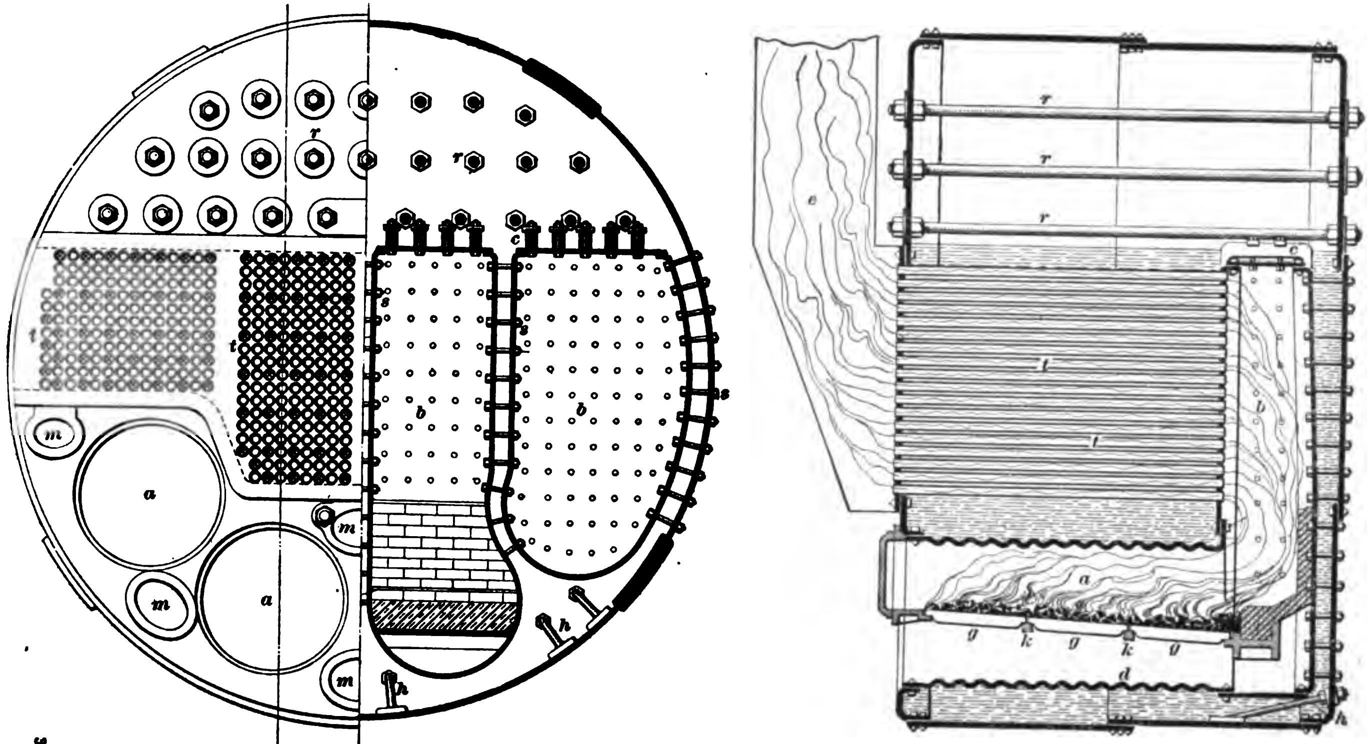


Fig. 18

The flat heads of the shell are kept from bulging by the heavy stayrods *r*, *r*, and by the diagonal braces *h*, *h*. About one-third of the tubes (those marked with a cross in the figure) are threaded and provided with nuts, and thus act as stayrods for the flat surfaces occupied by the tubes. The flat sides of the combustion chambers are stayed to each other and to the rear head by the staybolts *s*, *s*. The flat tops of the combustion chambers, called the **crown sheets**, are strengthened by the girder stays, or **crown bars**, *c*, *c*. The manholes and handholes *m*, *m* give access to the various parts of the boiler. The various fittings are not shown in the figure, but are attached in convenient places.

The furnaces are placed within the corrugated flues *a*, *a*. Since the width of the grates is limited to that of the corrugated flues, it is necessary to make the grates long in order to provide the required grate area. As shown, the grates *g* are made in three sections, supported by the cross-bars *k*, *k*. Below the grates are the ash-pits *d*. At the rear end of each grate is placed a firebrick plate for the purpose of preventing cold air from sweeping through the ash-pit into the combustion chamber without first passing through the grate.

The gases arising from the combustion of the coal pass into the combustion chambers *b*, where they undergo further combustion in contact with the air that passes through the grates. The hot products of combustion then pass through the tubes to the smokebox *e* and out through the stack.

It is seen from the figure that the flues and tubes are completely surrounded by water, as are the combustion chambers. It is evident that this type of boiler has a very large heating surface in proportion to its cubic contents.

24. Double-Ended Scotch Boilers.—Double-ended Scotch boilers have furnaces at each end, and resemble two single-ended boilers placed back to back. A double-ended boiler is lighter, cheaper, and occupies less space than two single-ended boilers. In double-ended Scotch boilers, the furnace flues at each end communicate with a centrally located combustion chamber, from which the products of combustion pass through fire-tubes, as in Fig. 18, leading to two smoke flues, one at each end. Sometimes the boiler is so arranged that each opposite pair of furnace flues opens into a common combustion chamber. In such a case, each combustion chamber will have two nests of tubes, one nest connecting it with one head, the other nest with the other head. The gases from two opposite furnaces mix together in the common combustion chamber, and then pass through the two nests of tubes, one-half to one smoke flue, the other half to the other. In other respects, the construction of the boiler is similar to that shown in Fig. 18.

25. On account of the high steam pressures used in modern marine engines, the marine boiler must be carefully designed for strength. It is likewise necessary to reduce its weight and size to the lowest possible limits. The following data relating to the boilers of a naval vessel will give an idea of the principal dimensions of a Scotch marine boiler:

Boilers of Siemens-Martin steel—

Diameter of shell	15 ft. 2 in.
Length of shell	9 ft. 6 in.
Working pressure	135 lb. per sq. in.
Thickness of shell plates	$1\frac{5}{32}$ in.
Thickness of heads	$\frac{7}{8}$ in.
Number of furnace flues	4
Diameter of furnace flues	3 ft. 1 in.
Thickness of furnace flues	$\frac{15}{32}$ in.
Diameter of stayrods	$1\frac{15}{16}$ in.
Diameter of staybolts	$1\frac{1}{4}$ in.
Number of tubes	490
Diameter of tubes	$2\frac{1}{2}$ in.
Length of tubes	6 ft. 8 in.
Heating surface	2,500 sq. ft.
Weight without water	about 40 tons

It is clear from the above that, for the sake of safety, the Scotch type of boiler must be made extremely heavy and bulky when high steam pressures are used, and much attention is

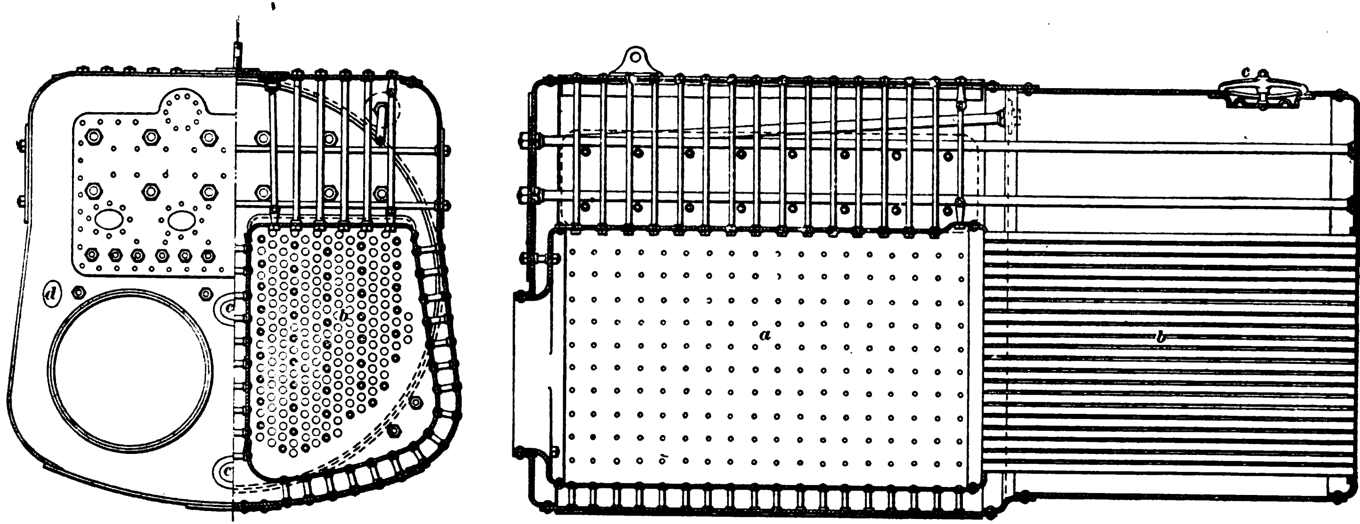


FIG. 19

being paid to devising a type of boiler that, while retaining the good features of the Scotch type, will be lighter, smaller, and cheaper for the same power.

26. Boilers for Gunboats.—In some gunboats and other small naval vessels, there is not sufficient room under the decks for the large Scotch boilers, and the type of boiler shown in Fig. 19, resembling the locomotive boiler, is frequently used. It is a plain cylindrical boiler with two rectangular fireboxes *a* (only one of which is shown), each connected by a nest of fire-tubes *b* to the rear boiler head. The furnaces are large, so as to leave sufficient space for combustion over the fires. Handholes *c* and *d* are located in the front head; and on top of the shell, near the rear end, is a manhole *c*, which affords ready access to the interior of the boiler for inspection, cleaning, or repairs.

WATER-TUBE MARINE BOILERS

27. Types of Water-Tube Marine Boilers.—The water-tube boilers in marine service resemble the water-tube boilers already described, and possess the same advantages and disadvantages. There are, however, two types: the small tube and the large tube. The small tube is lighter in weight and holds less water, but it is more difficult to clean; it is suitable for yachts and torpedo boats, while the large tube type, of which the Babcock & Wilcox marine boiler (a modified form of the boiler described in Art. **16**) is a well-known representative, is better adapted to large ships. Water-tube marine boilers are lighter per horsepower than Scotch boilers.

28. Belleville Boiler.—One form of the large-tube boiler, known as the **Belleville boiler**, is shown in Fig. 20. It consists of a number of nearly horizontal tiers of water tubes *a*, screwed or expanded at each end into return bends *b*, making a series of zigzag inclined tubes, beginning at the top of the furnace door and ending at the steam drum *c*, which is located above the tubes. There is a handhole in

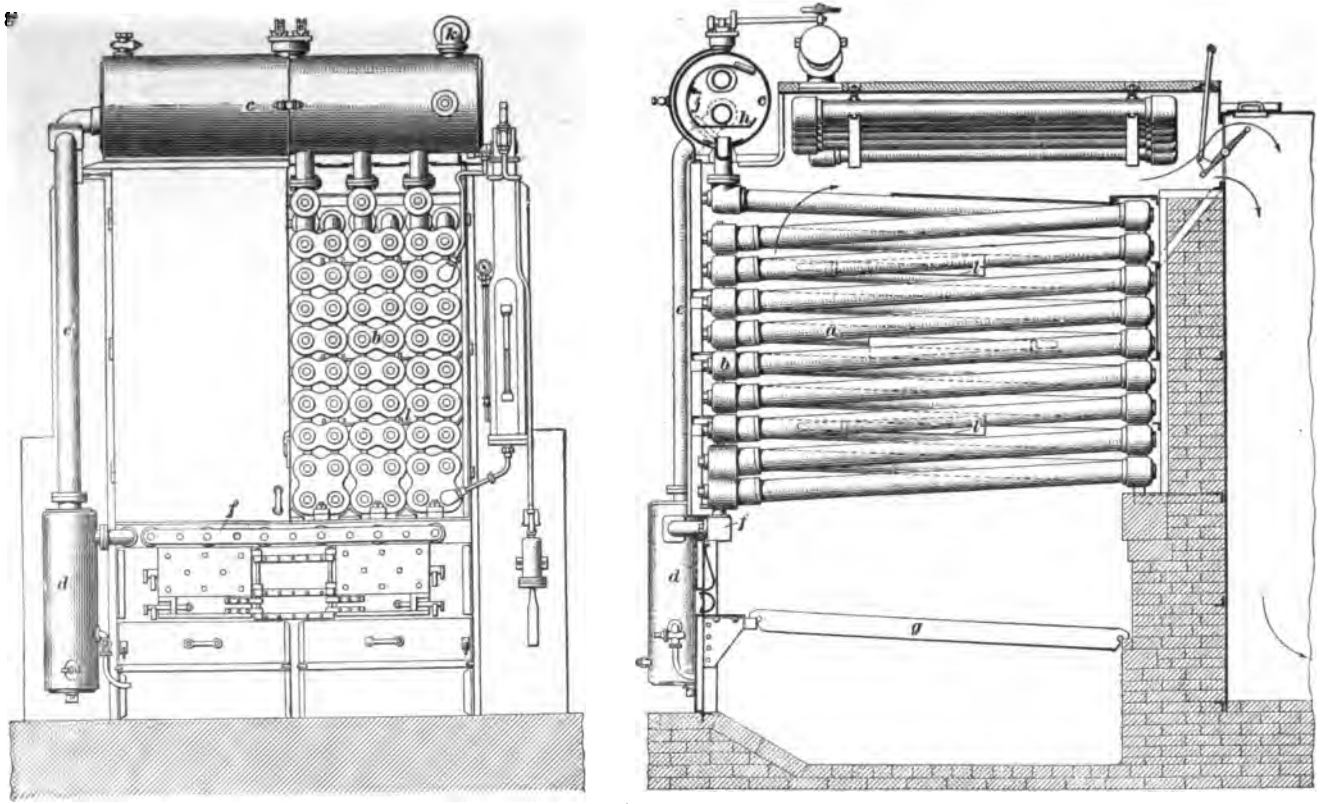


FIG. 20

each of the front bends or connecting boxes *b*. The mud-drum *d* stands vertically, and is located in front of the boiler and below the lowest tubes. The top of the mud-drum is connected to the bottom of the steam drum by a vertical pipe *e*. From the side of the mud-drum, a rectangular feed-pipe *f* extends across the front of the boiler, joining each vertical tier of water tubes *a*. The mud-drum has a handhole on the side near the bottom, and the blow-off is attached at the center of the lower head.

The boiler is enclosed in a steel casing. The fire-box is arranged below the tubes and runs their full length; the grate bars *g* slope downwards toward the rear. The products of combustion pass upwards between the tubes, thence about a superheater, and out near the top of the casing, as indicated by the arrows. Baffle plates *l* of steel or tile are fitted in the nest of tubes to deflect the hot gases, in order that the entire surface of the tubes may receive the benefit of the heat. The feedwater enters at one end of the steam drum and flows into a shallow pan *h*, then downwards through the external circulating pipe *e* to the mud-drum, and into the rectangular feedpipe *f*; thence it continues through the steam coils to the steam drum. The outlets of the water tubes in the steam drum are several inches above the bottom of the drum, so that the steam will not mingle with the comparatively cool water in the drum.

The water passes into the mud-drum through a non-return valve, and then to the bottom and up around a vertical baffle plate. The bottom of the drum forms a settling chamber, into which much of the sediment is deposited. The non-return valve keeps the water circulating in the same direction through the water tubes when the ship is rolling. It also regulates the direction of flow when steam is being raised. The casing of the boiler is made of steel plates riveted together. Angle irons are used at the joints for stiffeners. The upper part of the casing is lined with magnesia and asbestos, and the lower part next to the fire with firebrick.

This kind of boiler has very little water capacity, and hence it is usually fitted with an automatic feedwater regulator. In

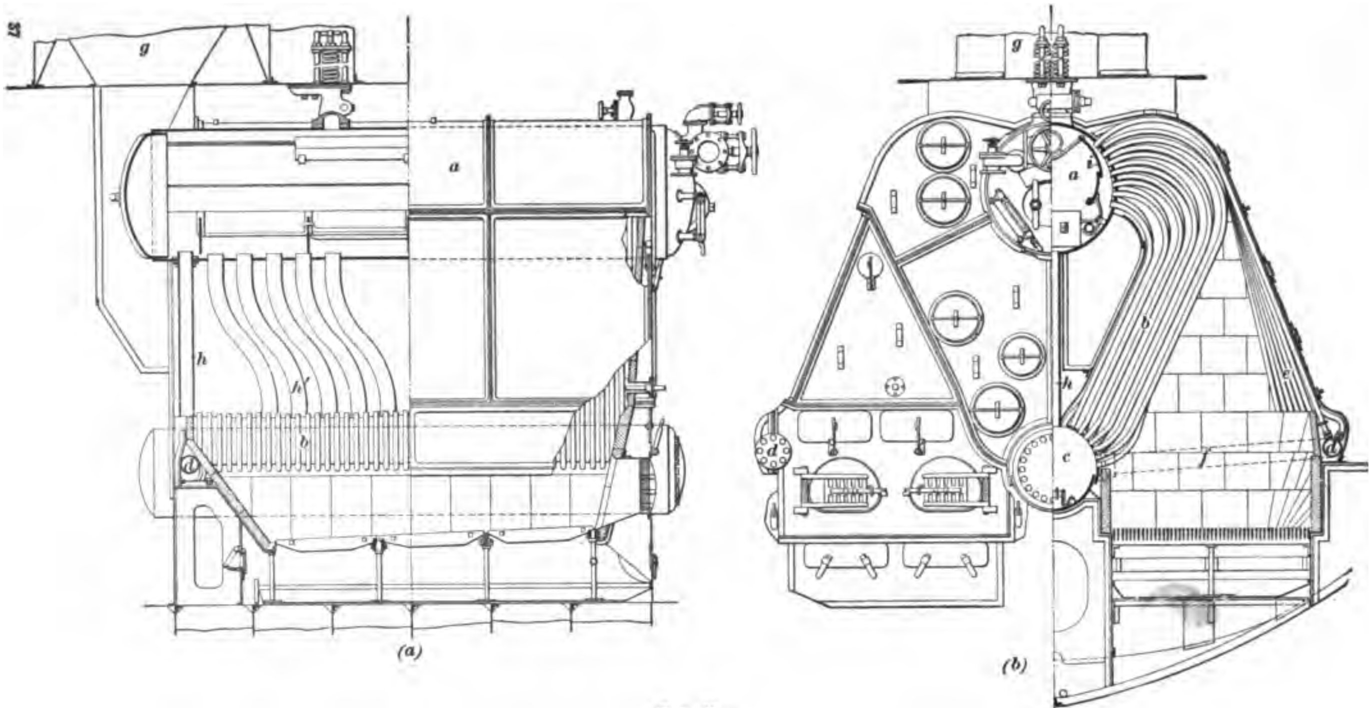


Fig. 21

operation, it requires very close attention. There is a strong upward flow of steam and hot water as they pass from the tubes into the steam drum. The pan *h* and its curved cover *j* serve as a deflector over the openings of the tubes to prevent the water from being carried out through the steam nozzle *k* on the top of the drum.

29. Thornycroft Boiler.—In Fig. 21 is shown a form of the **Thornycroft boiler,** a small-tube boiler much used on boats of very high speed. It consists of a large horizontal steam drum *a* at the top, connected by a series of bent tubes *b* to a small central drum *c* located at the bottom, between the furnaces. There are also two smaller drums *d*, *d*, at the outside edges of the grates. These side drums are connected by rows of bent tubes *e* to the steam drum *a*, and by a nearly horizontal pipe *f* to the lower central drum. There is a grate on each side of the central drum, and the products of combustion pass upwards between the tubes to the flue *g* at the front of the boiler. Inside the casing and near the front of the boiler are several large pipes *h h'*, known as *downcomers*, joining the steam drum *a* to the lower water drum *c*. The feedwater enters the steam drum and descends through the vertical downcomer *h* to the lower drum, a portion passing to the small side drums *d*, thence up through the bent tubes *b* and *e*, where the mingled steam and water is delivered against a baffle plate *i* inside the upper drum.

The boiler setting is made of a sheet-steel casing, lined with non-conducting material. Numerous doors are provided in the casing for cleaning and repairing the boiler. This type of boiler has been very highly developed and has proved very successful in torpedo-boat and torpedo-boat-destroyer service. Like all water-tube boilers, it holds very little water and is sensitive to slight changes in the condition of the fire.

30. Yarrow Boiler.—Another form of small-tube boiler, known as the **Yarrow boiler,** used in torpedo-boat service is shown in Fig. 22. It consists of a large steam drum *a*, with two smaller semicylindrical drums *b* below it and joined to it by inclined tubes *c*. The arrangement forms a triangle, with the grate *d* for the base. The lower drums have removable covers *e* for cleaning. The feedwater enters the steam drum below the water-line and descends through the inclined tubes most remote from the fire into the lower drum, deposits sediment, and rises through the tubes nearest the fire. The products of combustion pass between the tubes

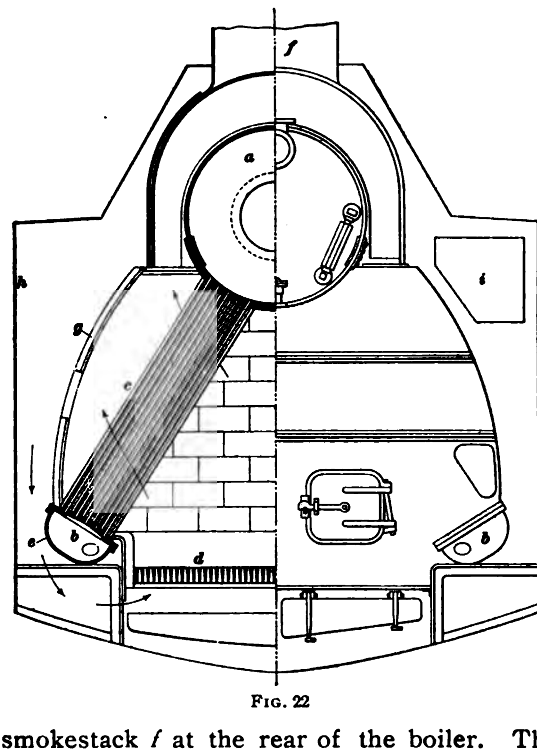


FIG. 22

to the smokestack *f* at the rear of the boiler. The boiler casing *g* is of iron and steel lined with non-conducting material. There is also an external casing *h*, so arranged that before entering the furnace the air for supporting combustion enters the opening *i* and flows between the casings *g* and *h*. This aids materially in keeping down the temperature of the boiler room by preventing the radiation of heat.

BOILER FITTINGS AND ACCESSORIES

BOILER FITTINGS

PROTECTIVE DEVICES

SAFETY VALVES

1. The **safety valve** is a device attached to the boiler to prevent the steam pressure from rising above a certain point. When steam is made more rapidly than it is used, its pressure must of necessity rise; and if no means of escape is provided for it, the result must be an explosion. Briefly described, the safety valve consists of a plate, or disk, fitting over a hole in the boiler shell and held to its place in one of three ways: (1) By a dead weight; (2) by a weight on a lever; (3) by a spring. The weight or spring is so adjusted that when the steam reaches the desired pressure the disk is raised from its seat, and the surplus steam escapes through the opening in the shell.

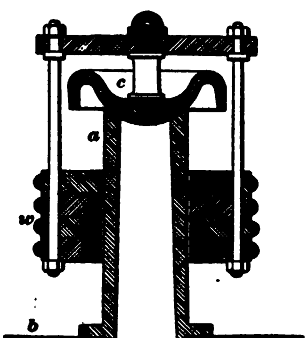


FIG. 1

2. In modern American practice, the **dead-weight safety valve** is used only for low-pressure boilers. It has the advantage of simplicity and compactness, and cannot readily be tampered with. The construction of a dead-weight safety valve is shown in Fig. 1. It consists of a hollow seat *a* attached to the boiler shell *b*, over which is fitted the valve disk *c*. The disk is loaded with a heavy weight *w* in the manner shown.

Let A = area of opening in valve seat, in square inches;
p = pressure at which valve is to blow off, in pounds per square inch;
W = dead weight, in pounds, including weight of valve.

As soon as the total pressure on the valve is slightly in excess of the dead weight, the valve is lifted and the steam escapes. Therefore, the dead weight must equal the pressure per square inch multiplied by the area of the opening; that is,

$$W = Ap$$

3. A **lever safety valve** is shown in Fig. 2. The steam from the boiler enters at *s* and escapes at *r* when the

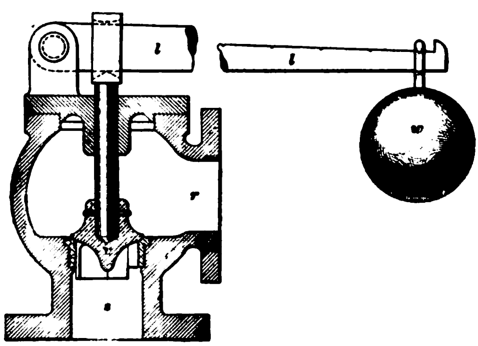


FIG. 2

steam pressure is sufficient to raise the valve *v* from its seat. The valve is held to its seat by the weight *w* hung from the lever *l*. The load on the valve is changed by shifting the weight along the lever. Notches are cut into the lever, and figures stamped below the notches indicate the blow-off pressures, in pounds per square inch, when the weight is hung under those figures.

4. A spring-loaded, or pop, safety valve is shown in Fig. 3. The valve *a a* rests on the two annular flat seats *b b* and *c c*, and is held down against the steam pressure by the spring *d*, the tension of which is regulated by the bolt *e*. The larger of the two seats *c c* is formed on the upper edge of the valve body; the smaller seat *b b* is formed on the upper edge of the cylindrical chamber *f*. This chamber is closed at the bottom and connects with the shell, or body, of the valve by means of the hollow arms *g*, *g*, the passages *h*, *h* allowing the steam to escape to the atmosphere when the valve is open.

Fig. 3

When the pressure under that part of valve *a* between the seats *b b* and *c c* reaches the maximum allowable pressure, the valve *a* lifts slightly and allows steam to escape past the seat *c c* into the space *i* inside the casing *j*, and through the holes *k* into the atmosphere. At the same time, steam passes seat *b b* into chamber *f*, thence through the passages *h*, *h* to the atmosphere. As the boiler pressure increases, the valve rises and steam flows into the chamber *f* faster than it can escape through the passages *h*, *h*, the pressure under valve *a* inside the inner seat *b b* thus quickly increases. This additional pressure overcomes the increasing resistance of the spring and forces the valve wide open, quickly relieving the boiler.

SAFETY-VALVE CALCULATIONS

5. Weight and Position of Ball.—In the diagrammatic representation of the safety-valve lever shown in Fig. 4, the

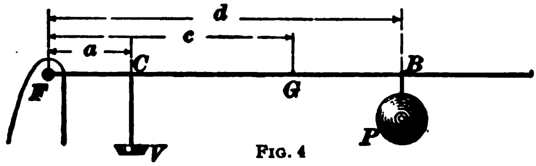


FIG. 4

valve stem and weight are attached to the lever at C and B, respectively; the fulcrum is at F.

Let $d = FB =$ distance from fulcrum to weight, in inches;
$c = FG =$ distance from fulcrum to center of gravity of lever, in inches;
$a = FC =$ distance from fulcrum to center line of valve, in inches;
$A =$ area of orifice beneath bottom of valve, in square inches;
$W =$ weight of ball P, in pounds;
$W_1 =$ weight of valve and stem, in pounds;
$W_2 =$ weight of lever, in pounds;
$p =$ blow-off pressure, in pounds per square inch.

It is a necessary condition of equilibrium that the algebraic sum of the moments of all the forces about a given point shall equal zero. Hence, taking F as the center of moments and treating all the forces acting downwards as positive and the upward pressure of the steam as negative, it follows that, since the upward steam pressure is pA,

$$Wd + W_1 a + W_2 c - pAa = 0, \text{ or}$$

$$W = \frac{a(pA - W_1) - W_2 c}{d} \qquad (1)$$

and

$$d = \frac{a(pA - W_1) - W_2 c}{W} \qquad (2)$$

EXAMPLE.—The area of the orifice is 10 square inches, the distance from the valve to the fulcrum is 3 inches, and the length of the lever is 32 inches. The valve and stem weigh 5 pounds, the lever weighs 12 pounds, and the gauge pressure is 90 pounds. What should be the weight W, if placed 2 inches from the end of the lever, assuming the lever to be straight?

SOLUTION.—Substituting in formula **1**, and since $c = \frac{32}{2} = 16$ in., and $d = 32 - 2 = 30$ in. for this case,

$$W = \frac{3(90 \times 10 - 5) - 12 \times 16}{30} = 83.1 \text{ lb. Ans.}$$

6. Having decided on the weight to be used, formula **2** of Art. **5** will give the distances from the fulcrum at which the weight may be placed, in order to allow the boiler to blow off at different steam pressures.

EXAMPLE.—Suppose all the quantities to remain the same as in the solution of the last example, except that it is desired that the boiler should blow off at 75 pounds gauge pressure instead of 90 pounds; what will be the distance of the weight from the fulcrum?

SOLUTION.—Applying formula **2** of Art. **5**,

$$d = \frac{3(75 \times 10 - 5) - 12 \times 16}{83.1} = 24.58 \text{ in. Ans.}$$

7. Area of Safety Valve.—By **area of safety valve** is meant the area of the opening in the valve seat or the area of the surface of the valve in contact with steam when the valve is closed. The area of the valve should be at least large enough to discharge steam as fast as the boiler can generate it, for otherwise the steam pressure would rise, even though the safety valve were open. Authorities differ greatly in their opinions regarding the area of valve that will be sufficient to fulfil this requirement.

The size of the valve relative to the size of boiler and working pressure is prescribed by law in many localities, and must be made to conform to the law wherever such law is in existence. In localities having no law governing this matter, the size of the safety valve may be calculated by the accompanying formulas, which are based on practice and recommended by leading authorities.

For natural draft,

$$A = \frac{22.5\,G}{p + 8.62} \qquad (1)$$

For artificial draft,

$$A = \frac{1.406\,w}{p + 8.62} \qquad (2)$$

in which G = grate surface, in square feet;
p = steam pressure, gauge, in pounds per square inch;
w = weight of coal burned per hour;
A = least area of safety valve, in square inches.

Professor Thurston proposed the formula,

$$A = \frac{.5\,W}{p + 10} \qquad (3)$$

in which W = weight of steam generated per hour, in pounds.

While these formulas are in common use, they give widely varying results. A comparison of the results obtained by using these formulas may be made by means of the following example:

EXAMPLE.—With a boiler having a grate surface of 30 square feet, burning 400 pounds of coal per hour, and generating 3,000 pounds of steam per hour at 100 pounds gauge pressure, what should be the area of the lever safety valve?

SOLUTION.—Applying formula **1**,

$$A = \frac{22.5 \times 30}{100 + 8.62} = 6.21 \text{ sq. in. Ans.}$$

By formula **3** the result given is

$$A = \frac{.5 \times 3{,}000}{100 + 10} = 13.64 \text{ sq. in. Ans.}$$

8. Area of Safety-Valve Opening.—Safety valves are often made with seats at an angle to the face of the valve, as shown in Fig. 5. The angle almost universally used is 45°.

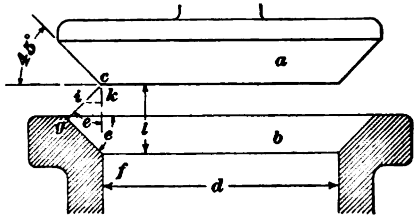


FIG. 5

When the valve disk is lifted from its seat, the annular space between the disk and the seat is called the **valve opening.**

In Fig. 5, the valve a is lifted vertically from its seat b a distance l, called the **lift** of the valve. From one corner c of the valve disk, a line cg is drawn perpendicular to the seat. The opening of the valve is a frustum of a cone with cg as the slant height. The area of the opening of the valve is then found by multiplying the slant height by one-half the sum of the circumferences of the upper and lower bases.

Let l = valve lift, in inches;
e = angle that seat makes with the face;
d = diameter of valve, in inches;
a = area of valve opening, in square inches.

The mean diameter between the upper and lower bases, which is taken at the point i, midway between c and g, equals

$$d + 2ki = d + 2ci \sin e = d + cg \sin e$$

The mean circumference is therefore $\pi(d + cg \sin e)$. This value multiplied by the slant height gives the area of opening, $a = cg \times \pi(d + cg \sin e)$.

But $cg = l \cos e$. Therefore,

$$a = 3.1416\, l \cos e(d + l \cos e \sin e) \qquad \textbf{(1)}$$

For an angle of 45° this formula reduces to

$$a = 2.221\, l\left(d + \frac{l}{2}\right) \qquad \textbf{(2)}$$

EXAMPLE 1.—What will be the area of the opening of a safety valve with a diameter of 5 inches and a seat angle of 45°, if the lift is .175 inch?

SOLUTION.—Using formula **1**,

$a = 3.1416 \times .175 \times .70711(5 + .175 \times .70711 \times .70711) = 1.977$ sq. in. Ans.

EXAMPLE 2.—What will be the area of the opening of a safety valve 4 inches in diameter with a vertical lift of .14 inch, the valve seat having an angle of 45°?

SOLUTION.—Applying formula **2**,

$$a = 2.221 \times .14\left(4 + \frac{.14}{2}\right) = 1.2655 \text{ sq. in. Ans.}$$

GRADUATION OF SAFETY-VALVE LEVER

9. The practical method of graduating a valve lever is to attach the valve to the lever and balance both over a knife edge; then, measure the distance from the point of suspension to the center of the pin, or fulcrum, on which the lever turns. Calling this distance b, and letting

$W_3 = W_1 + W_2$ in formula **2** of Art. **5,** then formula **2** may be written

$$d = \frac{p\,A\,a - W_3 b}{W}$$

To show that this formula will give the same results as formula **2** of Art. **5,** let all conditions remain the same as in the example given in Art. **6.** Imagine the lever and valve to be balanced on the knife edge, and take the center of moments at the balancing point. The weight of the lever per inch of length is $\frac{12}{32} = \frac{3}{8}$ pound. Consequently, applying the rule for moments

$$\frac{3}{8}(32 - b)\frac{32 - b}{2} - \frac{3}{8}b \times \frac{b}{2} - (b - 3) \times 5 = 0,$$

$$\text{or } b = 12\tfrac{3}{17} \text{ inches}$$

Substituting this value of b in the formula just given

$$d = \frac{75 \times 10 \times 3 - 17 \times 12\tfrac{3}{17}}{83.1} = 24.58 \text{ inches,}$$

the same as before.

The foregoing methods will not give exact results in practice, owing to the slight friction of the moving parts. An exact method for graduating a safety-valve lever, or of ascertaining whether a safety-valve lever has been graduated correctly or not, is the following, in which an ordinary platform scale may be used:

A 3-inch safety valve has been set to blow off at 100 pounds per square inch, but fails to do so when the steam gauge indicates 100. To ascertain if the lever has been graduated correctly, proceed as follows: Remove the valve and cover, and bolt the cover *E* to a couple of short, heavy timbers *A* and *B* suspended above the platform of the scale, as shown in Fig. 6.

The timbers *A* and *B* should be bolted to the floor to prevent them from tipping over. Now, adjust the height of the valve *C* so that the lever *D* will be horizontal. If the lever is too low, slip pieces of sheet iron or other metal under the rollers of the scale until the lever is horizontal; if too high, slip pieces under the cover *E*. Having placed the lever in a horizontal position, place the weight *W* at the

100-pound notch. The diameter of the valve being 3 inches, its area is $3^2 \times .7854 = 7.0686$ square inches. The total steam pressure necessary to raise the valve when the weight *W* is at the 100-pound notch is $7.0686 \times 100 = 706.86$ pounds, say 707 pounds. If the scale balances when set for 707 pounds, the lever has been graduated correctly. If it does not balance

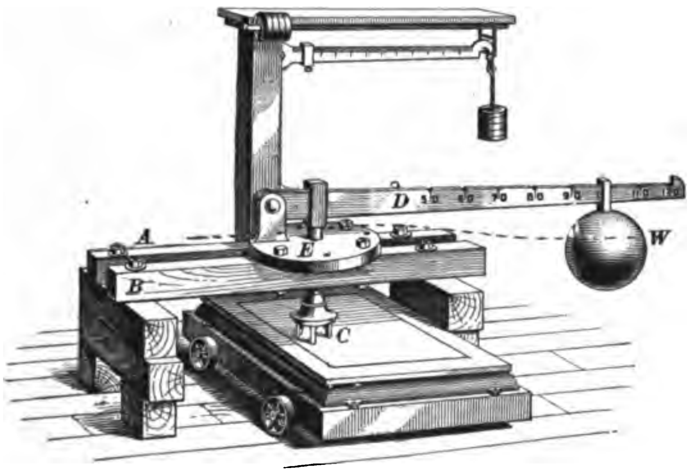


Fig. 6

at this point, shift the weight slightly to the right or left until the scale balances; mark this point on the lever; it will be the correct 100-pound notch. In the same way, test the graduation of the other points of the scale, for example, the 80-pound notch. For this point, the scale should balance at $7.0686 \times 80 = 565.488$ pounds, say 565.5 pounds.

LOCATION OF SAFETY VALVE

10. The safety valve should be placed in direct connection with the boiler, so that there can be no possible chance of cutting off the communication between them. A stop-valve placed between the boiler and safety valve is a very fruitful cause of boiler explosions. Again, the safety valve must be free to act, and to prevent it from corroding fast to its seat, it should be lifted from the seat occasionally.

Care must be taken to prevent ignorant persons from raising the blow-off pressure by adding to the weights or increasing the tension of the spring. To this end, the weights of lever safety valves are often locked in position by the boiler inspector.

EXAMPLES FOR PRACTICE

1. A boiler generates 1,400 pounds of steam per hour at a pressure of 85 pounds per square inch. (*a*) What should be the diameter of the safety valve opening? (*b*) Supposing the valve to be of the dead-weight type, what should be the weight? Ans. $\begin{cases}(a)\ 3\frac{1}{16}\text{ in.}\\(b)\ 626\text{ lb.}\end{cases}$

2. The diameter of the valve opening is $4\frac{3}{8}$ inches, the length of the lever is 35 inches, the distance from fulcrum to valve stem is $3\frac{1}{2}$ inches, and the steam pressure is 80 pounds. The weight of the lever is 16 pounds and of the valve 6 pounds. How far from the fulcrum should a weight of 130 pounds be placed? Ans. 30 in.

3. If the weight in example 2 were hung at the end of the lever, what would be the blow-off pressure? Ans. 92.2 lb.

4. A boiler evaporates 3,500 pounds of water per hour and generates steam at an average pressure of 95 pounds; what should be the diameter of the safety-valve opening? Ans. $4\frac{3}{8}$ in., nearly

5. Assuming the safety valve of example 4 to be of the lever type, what weight should be placed 40 inches from the fulcrum, the valve being 4 inches from the fulcrum? Neglect the weight of valve and lever. Ans. 158.3 lb.

6. What is the area of the opening of a 4-inch safety valve with a lift of $\frac{3}{16}$ inch, if the valve seat is at 45° to the face? Ans. 1.7 sq. in.

FUSIBLE PLUGS

11. Fusible plugs are devices placed in the crown sheets of furnaces to obviate danger from overheating through lack of water. The plug consists of an alloy of tin, lead, and bismuth, which melts at a comparatively low temperature. So long as the furnace crown is well covered with water, the plug is kept from melting by the comparative coolness of the water, but should the water sink low enough to uncover the top of the plug, the fusible part quickly melts and allows the steam and water to rush into the furnace, and thus give warning of low water. In many localities, the law requires that fusible plugs shall be attached to all high-pressure boilers.

The fusible plugs in common use are shown in section in Fig. 7. They consist of brass or iron shells threaded on the outside with a standard pipe thread. The plugs have some form of conical filling, the larger end of the filling receiving the steam pressure. The conical form of the filling prevents it from being blown out by the pressure of the steam.

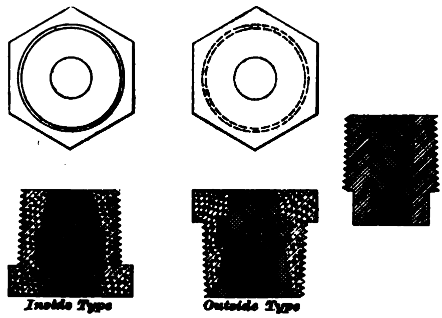


Fig. 7

12. A form of plug especially adapted to internally fired boilers of the locomotive type is shown in Fig. 8. The plug *a* is screwed into the crown sheet *b*, and the fusible cap *c* is laid on top of it and kept in place by the nut *d*. A very thin copper cup *e* is placed over the top of the cap *c* to protect it from any chemical action of the water. The top of the cap extends from $1\frac{1}{2}$ to 2 inches above the crown sheet, so that when it melts, on account of the water being too low, there will still be enough water left to protect the sheet from being overheated, or *burned*, as it is often called.

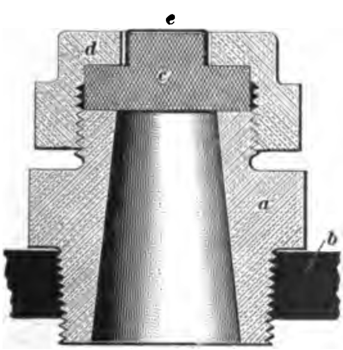


Fig. 8

13. In horizontal return-tubular boilers, the plug is usually placed in the back head 3 inches above the upper row of tubes. In flue boilers of the two-flue type, one plug is screwed in each flue at its highest point, or in the back head, about 2 or 3 inches above the tops of the flues. The latter practice is considered the better, since it will give warning of lack of water before the flues become uncovered. In firebox boilers, the plug is screwed into the highest point of the crown sheet. In vertical boilers, it is usually screwed into one of the tubes about 2 inches below the lowest gauge-cock. In water-tube boilers, it is usually located in the shell of the steam drum. In general, it should be so located that it will prevent, by the warning it gives, the overheating of the parts within the fire-line.

HIGH- AND LOW-WATER ALARMS

14. A device is often attached to the boiler to give an audible warning, usually by blowing a whistle, of a shortage or a surplus of water. Devices that indicate a shortage of water are called **low-water alarms;** those that indicate both a surplus and a shortage of water are called **high- and low-water alarms.**

In purely low-water alarms, the whistle may be sounded by the melting of a fusible plug, which, through the falling of the water level in a separate chamber outside of the boiler, is brought in contact with the steam. Fusible-plug alarms are cheap and easily applied; they are rather unreliable, however, because they are liable to become incrusted with scale.

Most low-water alarms employ a float operating a valve leading to a steam whistle, the float being buoyed up by the water. Their construction is similar to that of high- and low-water alarms.

15. One form of high- and low-water alarm is shown in Fig. 9. The device consists of two hollow floats *a*, *b* suspended from the bell-cranks *c*, *d*. To the short arm of each bell-crank is attached the valve stem of a small valve, there being one valve for each float. These valves serve to put the steam space of the water column in communication with the alarm whistle *e*. In this particular design, a sediment chamber *f* is formed at the bottom of the column and collects all foreign matter that settles from the water. The water-column drain is connected to the settling chamber. When the water is at its proper level, the float *b* is surrounded by water and, being hollow, is pressed upwards; this keeps the upper whistle valve closed. When the water becomes so low in the column as to begin to uncover the float, the upward pressure due to the buoyant effect of the water gradually diminishes, and finally becomes so small that the float descends, thus opening the upper whistle valve and sounding the alarm. The high-water alarm float *a* keeps the lower whistle valve closed by the weight of the float. When the water rises, the float is carried upwards, the lower whistle valve is opened, and the alarm sounded.

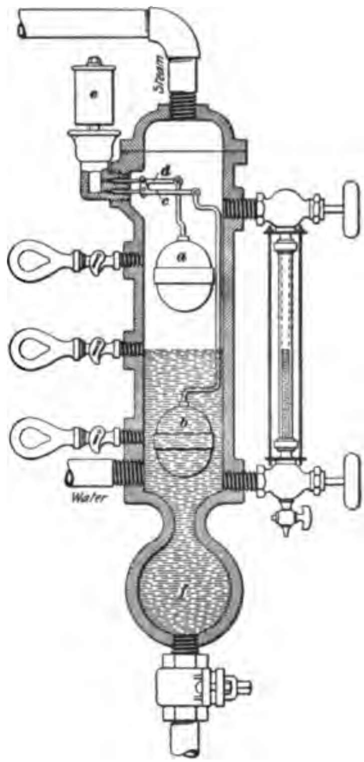


FIG. 9

Low-water alarms depending on the difference in expansion of different metals for actuating a whistle valve or electric bell have been used occasionally, and are on the market; they have not found much favor, however, chiefly because they are too delicate.

PRESSURE AND WATER LEVEL INDICATORS

STEAM GAUGE

16. The **steam gauge** indicates the pressure of the steam contained in the boiler. The most common form is the *Bourdon pressure gauge*, the distinguishing feature of which is a bent elliptic tube tending to straighten out under an internal pressure. Bourdon pressure gauges are made in various ways by different manufacturers; a very common design is shown in Fig. 10. It consists of a tube *a*, of elliptic cross-section, that is filled with water and connected at *b* with a pipe leading to the boiler. The two ends *c* are closed and are attached to a lever *d*, which is in turn connected with a quadrant *e*; this quadrant gears with a pinion *f* on the axis of the index pointer *g*. When the water contained in the elliptic tube is subjected to pressure, the tube tends to take a circular form and straighten out, throwing

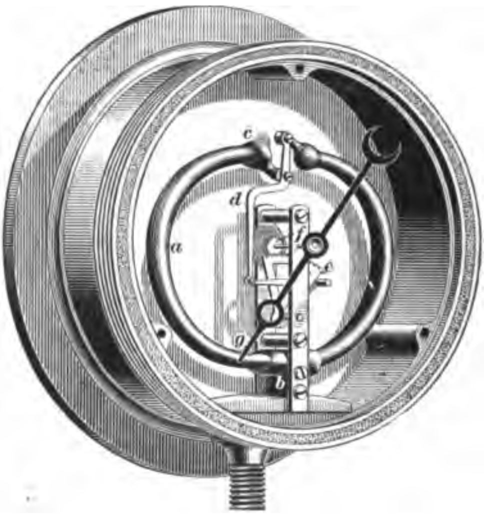


FIG. 10

out the free ends a distance proportional to the pressure. The movement of the free ends is transmitted to the pointer by the lever, rack, and pinion, and the pressure is thus recorded on a graduated dial in front, which has been removed in order to show the mechanism.

Pressure gauges for indicating steam pressure are invariably graduated to indicate pressure in pounds per square inch, wherever the English system of weights and measures is used, and show how much the pressure has been increased above the atmospheric pressure.

17. A steam-pressure recording gauge is a device that produces an accurate and complete record of the steam pressure, giving all the variations in pressure with their exact time and duration. A pencil or a pen is attached by means of suitable levers to a small piston acted on by the boiler pressure. The movement of the piston is registered on a chart by a pencil or pen under the tension of a spring so graduated that the pencil moves vertically 1 inch for each 100 pounds gauge pressure. The chart is moved horizontally by means of a clock that unwinds the chart from one drum and winds it on another, so that the horizontal graduations of the chart are in hours and fractional parts of an hour. A bell is arranged to sound an alarm should the pressure become too low or too high.

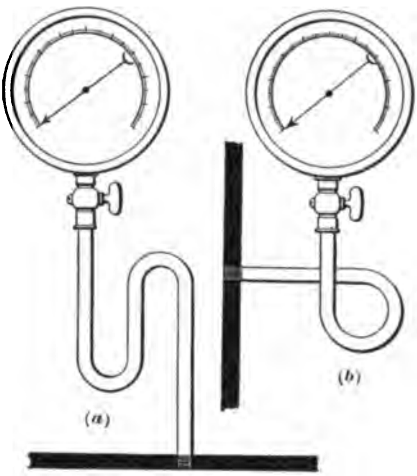


FIG. 11

With another form of recording gauge, a circular chart is used, the record of a whole day thus being visible. The charts are filed for reference purposes.

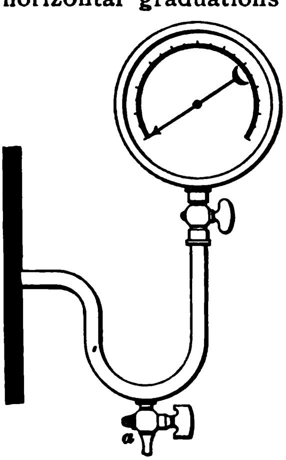


FIG. 12

18. A steam gauge should be connected to the boiler in such a manner that it will neither be injured by heat nor indicate incorrectly the pressure to which it is subjected. To prevent injury from heat, a so-called siphon, which may be made as shown in Fig. 11, (*a*) or (*b*), is usually placed between the gauge and the boiler. This siphon in a short time becomes filled with water of condensation that protects the spring of the gauge from the injury the hot steam would cause. Care should be taken not to locate the steam-gauge pipe near the main steam outlet of the boiler, since this may cause the gauge to indicate a lower pressure than really exists. In locating the steam gauge, care must also be taken not to run the connecting pipe in such a manner that the accumulation of water in it will cause an extra pressure to be shown.

The gauge connections shown in Fig. 11 cannot be drained without disconnecting them. To meet this drawback, the gauge may be provided with a petcock *a*, placed at the lowest point, as shown in Fig. 12.

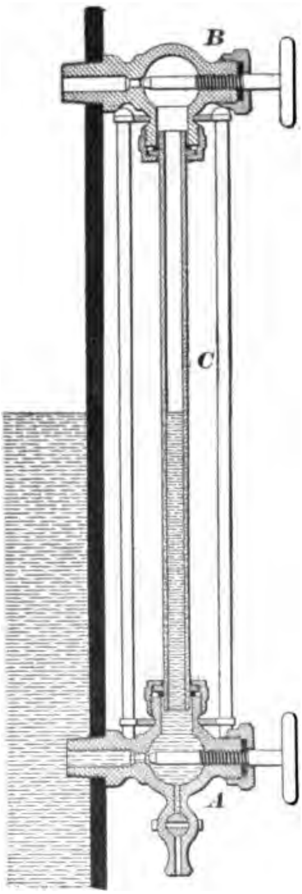


Fig. 13

GLASS WATER GAUGES

19. The **gauge glass** is a glass tube whose lower end communicates with the water space of the boiler and whose upper end is in communication with the steam space. Hence, the level of the water in the gauge should be the same as in the boiler.

Fig. 13 shows a common form of gauge-glass connection. The lower fitting *A* connects with the water space of the boiler, and the upper fitting *B* with the steam space. A drip cock is placed at the lower end of the glass for the purpose of draining it. Two or more brass rods are provided to protect the gauge glass against accidental breakage. The fittings may be screwed directly into the boiler. The gauge should be so located that the water will show in the middle of the gauge glass, as at *C*, when at its proper level in the boiler. Both fittings are provided with valves, by means of which communication with the boiler can be shut off in case the gauge glass breaks.

20. To prevent loss of steam and water and to obviate the danger of scalding the hands and face in shutting off the valves, it is desirable to have water gauges that will automatically shut off communication with the boiler whenever the gauge glass breaks. There are many designs of such water gauges on the market, one of which, typical of the others, is shown in Fig. 14. A ball is placed within the shank of each fitting, as shown, and is prevented from falling out by a brass pin. Should the gauge glass break, the out-rushing steam and water carry the balls forwards and thus close the openings leading to the gauge glass. While the balls may not shut off the steam and water entirely, they will check the outflow sufficiently to permit the valves to be closed without danger.

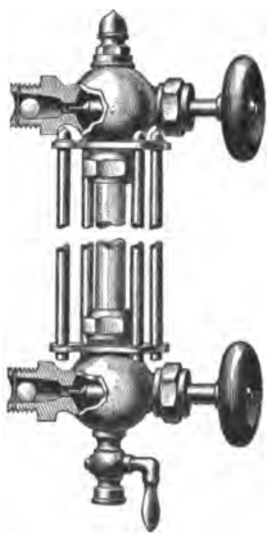


FIG. 14

To obviate the danger of scalding the hands, a two-armed lever is sometimes placed on each valve stem. Chains are led from the ends of the levers to some safe point, and the valves are opened and closed by pulling the chains.

21. Glass water gauges connected directly to the boiler are open to the objection that the violent ebullition at the surface of the water will cause them to indicate a wrong water level. To overcome this objection, they are frequently placed on a separate fitting, as in Fig. 15, known as a **water column**, which consists of a large hollow tube with its ends connecting with the steam and water spaces of the boiler far enough above and below the water level to be out of reach of the violent ebullition of the water at its surface, and thus insuring correct gauge readings.

GAUGE-COCKS

22. A **gauge-cock** is a simple cock or valve attached either directly to the boiler or, preferably, to a water column for the purpose of testing the level of the water in the boiler. Three gauge-cocks are generally employed. The lowest is placed at the lowest level that the water may safely attain, and the uppermost at the highest desirable level. The third cock is placed midway between the other two. On opening a cock above the water level, steam will come out, and on opening one below the water level, water will appear. Hence, the water level may easily be located by opening the cocks in succession.

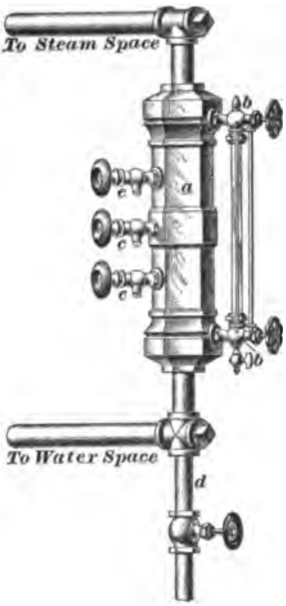


FIG. 15

23. Gauge-cocks are commonly of the type shown in connection with the water columns illustrated in Figs. 15 and 16. Such cocks provided with wooden hand wheels consist of brass bodies having threaded shanks for attaching them to the boiler or water column. The seat within the body of the cock is closed by the end of the threaded valve stem. The steam or water issues from the nozzle when the cock is open. Gauge-cocks can be obtained with a lever handle in the form of a crank. Such cocks can be operated from a distance by means of a rod. In some designs, the valve is held to its seat by a strong spring that automatically closes the valve the moment the hand releases it.

24. Weighted gauge cocks are shown in connection with the water column illustrated in Fig. 9. A weighted cock consists of a body having a threaded shank for attaching it to the boiler or water column. The weight is pivoted to the body, and when down presses a strip of rubber packing against the face of the opening of the cocks. The packing must be renewed quite frequently, as it rots under the high temperature to which it is subjected. The cock is opened by lifting the weight slightly, and the issuing steam or water is deflected downwards by the curved end wall of the slot.

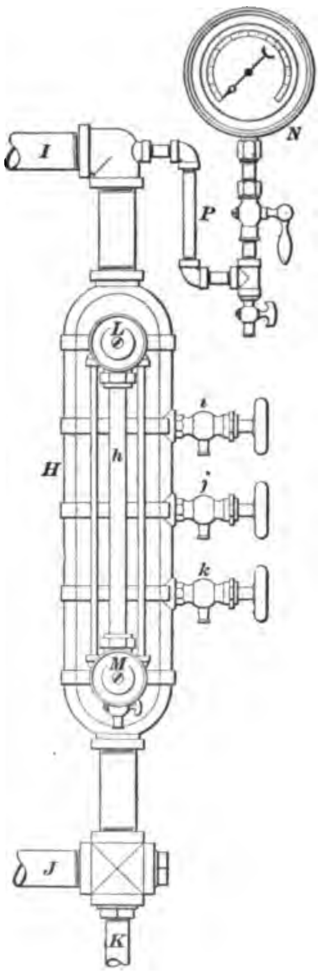


FIG. 16

WATER COLUMNS

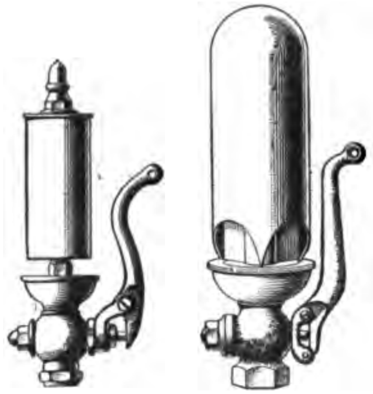
25. A common form of **water column** is shown in Fig. 15. It consists of a hexagonal cast-iron stand pipe *a* tapped on top and bottom for pipe connections to the boiler. Tapped bosses are provided to receive the threaded shanks of the gauge-glass fittings *b*, *b* and the gauge-cocks *c*, *c*, *c*. Each maker has his own style of stand pipe, the different makes varying chiefly in the ornamentation. The steam gauge is frequently mounted on top of the water column.

26. The connection to the boiler should be made with a **T** on top and a cross on the bottom, as shown, plugging up the unused openings with brass plugs. If the connections are made in this manner, they can be cleaned with a rod when the plugs are unscrewed. A drain pipe *d* with a valve in it, and leading to the ash-pit, should always be provided, and should be used frequently for blowing out sediment collecting in the stand pipe. For low-pressure boilers, no valves are required in the pipes leading to the steam and water spaces of the boiler; for high-pressure boilers, however, valves should always be provided. These valves are used in blowing out the stand pipe and connections. Closing the valve in the upper pipe and opening the valve in the drain pipe blows out the lower pipe; closing the valve in the lower pipe and opening the valve in the drain pipe blows out the upper pipe and the stand pipe.

27. Fig. 16 illustrates how a steam gauge *N* may be attached to the water column by means of a siphon *P* made from ordinary brass pipes and fittings. The stand pipe *H* carries on one side the upper fitting *L* and lower fitting *M* for the gauge glass *h*, and at right angles to this gauge the three gauge-cocks *i*, *j*, and *k*. The pipe *I* leads to the steam space of the boiler and the pipe *J* to the water space. A drain pipe *K* is attached to the cross as shown

STEAM WHISTLE

28. While the **steam whistle** is not essentially a boiler fitting, yet the fact that it is used in connection with high- and low-water alarms, besides being used for signaling purposes, warrants a description of its principle of action. Two of the most common constructions, as used for signaling, are shown in Fig. 17 (*a*) and (*b*). The bell, as shown in Fig. 17 (*a*), is a hollow cylinder closed at the top and open at the bottom, and is held in position by a stud that passes through the center and is secured at the upper end by means of a screw and jam nut. The hollow base has a narrow circular orifice that communicates with the steam pipe and valve. As the steam rushes out of the orifice in an upward direction, toward the mouth of the bell, it slightly compresses the air contained in the bell. The air being elastic will not remain compressed, but will spring back slightly toward the inrushing steam, to be again forced back, as before, thus causing vibrations. These vibrations continue so long as steam is permitted to flow, and are communicated to the surrounding atmosphere, thus producing sound.



(a) Fig. 17 (b)

29. The tone may be changed to a higher pitch by lowering, or to a lower pitch by raising, the bell. This may be done by loosening the jam nut and turning the bell up or down, after which the nut should be tightened again.

Whistles are also constructed to produce two or more tones of different pitch simultaneously by dividing the bell into two or more cell-like parts, as shown in Fig. 17 (*b*). Each compartment produces a different tone, and when these tones chord perfectly, the effect is quite pleasing.

30. In manufacturing establishments, the whistle for signaling is usually located on the roof; that is, at a considerable distance above the boiler. In order to prevent this long pipe from becoming filled with water, it is advisable to fit a small drain pipe and valve directly above the stop-valve in the whistle pipe, which is placed close to the boiler. At night, the steam may be shut off from the whistle and the drain valve opened.

BOILER ACCESSORIES

AUXILIARY APPLIANCES

STEAM DOMES

31. A **steam dome** is a cylindrical vessel riveted to the shell of horizontal power boilers for the purpose of increasing the steam space, and also for the purpose of drying the steam. The hole cut in the shell to give communication between the boiler and dome should be made only large enough to allow a man to pass through, since a large hole materially weakens the shell. The edge of the plate around the hole should be reenforced by a wrought-iron ring riveted to it. Small holes should be drilled through the shell plate at each side of the boiler inside of the dome to allow the water that accumulates there to drain back into the boiler.

The dome flange fitting the boiler shell is called the **saddle**, and should be double-riveted to the shell. Sometimes the diameter of the opening cut in the boiler shell is made the same as the internal diameter of the steam dome, the base of which is flanged outwardly and curved to fit the boiler shell, an internal reenforcing ring being employed to counteract the weakening effect of cutting a large steam dome opening in the shell. Steam domes usually have a diameter equal to one-half the diameter of the boiler, and a height equal to about nine-sixteenths the diameter of the boiler.

The top of the steam dome is closed by the **dome head**, which formerly was made of cast iron. Owing to the high pressures now carried, the use of cast-iron dome heads for high-pressure boilers has been almost entirely abandoned, because of the treacherous nature of the material. In the best modern practice, flanged and crowned steel heads are used, the head being crowned to a radius sufficient to make it stiff enough to withstand the pressure without additional bracing. When flanged flat steel heads are used, they must be well braced by diagonal braces.

STEAM DRUMS

32. Boilers are sometimes fitted with a **steam drum** instead of a dome. The steam drum is simply a cylindrical vessel connected to the shell. When several boilers are set so as to form a battery, they are often connected to a drum common to all boilers. When each boiler has its own furnace, there should be a stop-valve between each boiler and the drum to allow the boiler to be taken out of service when required. When the boilers in a battery have a furnace common to all, no stop-valve should ever be placed in the pipe connections between each boiler and the drum. Where boilers are in a battery with separate furnaces, each boiler must have its own safety valve, which should always be so fitted that it cannot be cut off from the boiler under any circumstances.

Longitudinal steam drums are sometimes attached to the boiler by two nozzles. This practice is objectionable, however, since with an unequal expansion of the boiler and drum, which is quite likely to occur, the joints of the nozzles will become leaky, owing to the stresses to which they are subjected. It is now the rule in good work to use one nozzle only. When the steam drum is used for a single boiler, its diameter is generally made about one-half the diameter of the boiler, and its length about the diameter of the boiler. Where one steam drum is common to several boilers, its diameter is made about one-half that of the boilers to which it is attached and its length about the horizontal outside-to-outside measurement over the several boiler shells. Steam drums require just as rigid inspection as the boiler itself.

MUD-DRUMS

33. A **mud-drum** is a cylindrical vessel occasionally attached to a boiler for the purpose of providing a place for the collection of mud and sediment from the feedwater. It is located beneath the boiler and at the rear end, being connected to the boiler by a suitable nozzle, usually of cast iron. When a mud-drum is provided, the blow-off should be attached to it and the sediment collected in the drum frequently blown out. Handholes are generally provided in the heads of mud-drums for convenience in removing the scale and sediment that fail to pass out through the blow-off.

MANHOLES AND HANDHOLES

34. Manholes.—For the purpose of allowing the inside of the boiler to be examined, cleaned, and repaired, holes closed by suitable covers are cut into the head or shell.

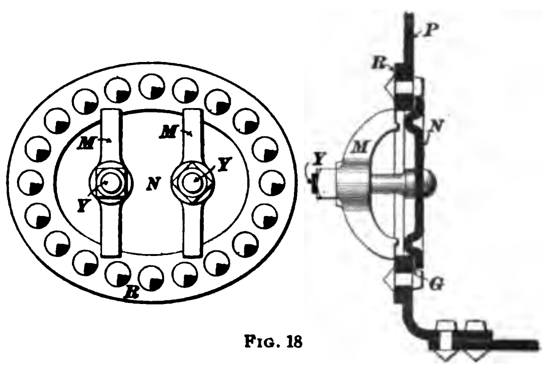


FIG. 18

When of sufficient size to admit a man, they are called **manholes**; otherwise, **handholes.**

A common construction of a manhole and its cover is shown in Fig. 18. An elliptic hole is cut into the head or shell of the boiler. The hole is made elliptic in shape to permit the cover to be passed through the hole, the short axes of which vary from 9 to 16 inches, while the long axes are usually from 15 to 18 inches. A wrought-iron or steel ring *R*, called a **compensating,** or **reenforcing, ring,** is riveted to the plate *P*, generally on the outside, for the purpose of strengthening the plate, which is weakened considerably by cutting such a large hole through it. A cover *N* made of wrought iron, cast iron, or steel is fitted to the hole, inside of the boiler, and is provided with two studs *Y*, *Y* riveted to it. This cover is flanged and overlaps the edges of the plate about 1 inch or more all around its perimeter. A yoke *M* is slipped over each stud, its two extremities resting on the compensating, or reenforcing, ring. A ring *G*, or **gasket,** as it is commonly called, made of sheet rubber or any other pliable waterproof material, is placed between the plate and the cover and serves to make a water-tight joint.

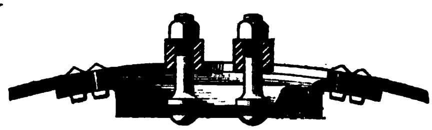


Fig. 19

Of late years, it has become quite generally the practice to flange the head inwards and face its edge, thus doing away with the necessity for the compensating, or reenforcing, ring. When the manhole is in the shell, in the best modern practice, a flanged compensating, or reenforcing, ring is riveted to the inside of the shell, as shown in Fig. 19. Manholes are usually made with inside dimensions of about 11 inches by 15 inches; if they are smaller, it is rather difficult for a man to get through them.

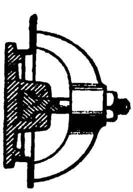


Fig. 20

35. Handholes are placed in boilers whose construction does not permit the entrance of a man, as, for example, in vertical boilers. They are also placed in other boilers in convenient positions: in boilers of the locomotive type they are usually placed in the corners of the water legs; and in horizontal return-tubular boilers, they are often found in the heads below the tubes. The handhole is a convenient place to rake out sediment and scale and to admit a hose for the purpose of washing out the boiler. The handhole and its cover are constructed very much like a manhole and cover as indicated in Fig. 20, the handhole being smaller, usually about 4 inches by 6 inches; but one yoke and bolt is required to close up the cover.

STEAM-DRYING DEVICES

36. At present, there is a tendency to discard steam domes and steam drums, installing slightly larger boilers in order to get the steam space, and fitting inside of them some form of dry pipe to insure dry steam. A **dry pipe** is a device for removing moisture from the steam before it leaves the boiler.

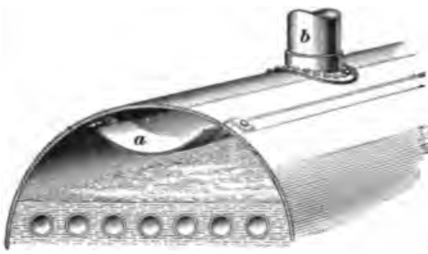


FIG. 21

37. One of the simplest forms of dry pipe is shown in Fig. 21. It is not strictly a dry pipe, but a baffle plate, or trough, *a*, made of light sheet iron, and extending from the front to the rear head of the boiler. The steam rising from the surface of the water is forced by the trough to change its

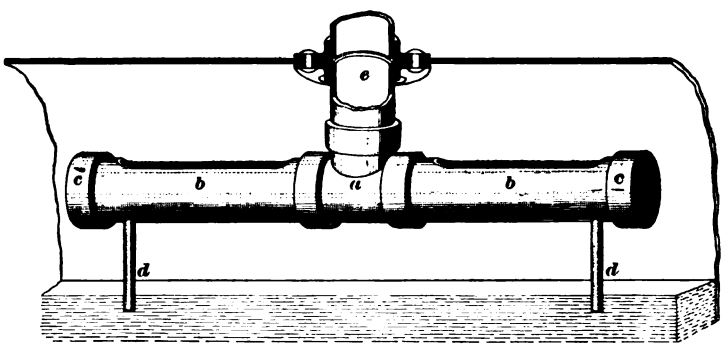


FIG. 22

direction of flow before entering the outlet pipe *b*, and is supposed to deposit its moisture in the trough, whence it drains back into the water. While this drying device undoubtedly takes out some moisture if the steam is very wet, it cannot be claimed to be particularly efficient. It is better, however, than no provision for drying.

38. A much-used and quite efficient and simple form of dry pipe is shown in Fig. 22. It is made from a **T** *a*, which is attached to the steam outlet by a nipple, and two pipes *b*, *b* slotted on top and having their ends closed by caps *c*, *c*. The steam enters through the slots on top, and, striking against the bottom of the pipe, deposits the water globules. The water collecting in the dry pipe drains back into the water space through drip pipes *d*, *d*. The combined area of the slots should be about one-third greater than the area of the outlet pipe *e*.

SUPERHEATERS

39. The demand for greater economy in the performance of steam engines has led to the development of the boiler accessory known as the **superheater**, by means of which the steam may be superheated so that it will contain more heat than would the same weight of saturated steam, and thus insure increased engine economy. In order to superheat the steam, it must pass from the boiler into a separate compartment and have more heat applied to it. This may be done with a separate furnace or by using a coil of pipe within the boiler setting itself.

In Fig. 23 (*a*) and (*b*) is shown one form of superheater as arranged in connection with a water-tube boiler. It consists of a number of bent tubes *a* with their ends expanded into headers *b*, *b'*, and is located in the upper part of the combustion chamber of the boiler. The upper header *b* is connected with the dry pipe *d* by two vertical pipes *c*, *c'*, while the lower header *b'* is connected by means of two pipes *e*, *e'* to the steam outlet *f* on top of the boiler. The steam is drawn from the dry pipe through the pipes *c*, *c'* to the upper header *b*, thence through the superheater tubes *a* to the lower header *b'*, and up the external pipes *e*, *e'*, to the steam outlet *f*. The lower header *b'* is connected to the water space of the boiler by means of the pipes *g* and *h*, fitted with

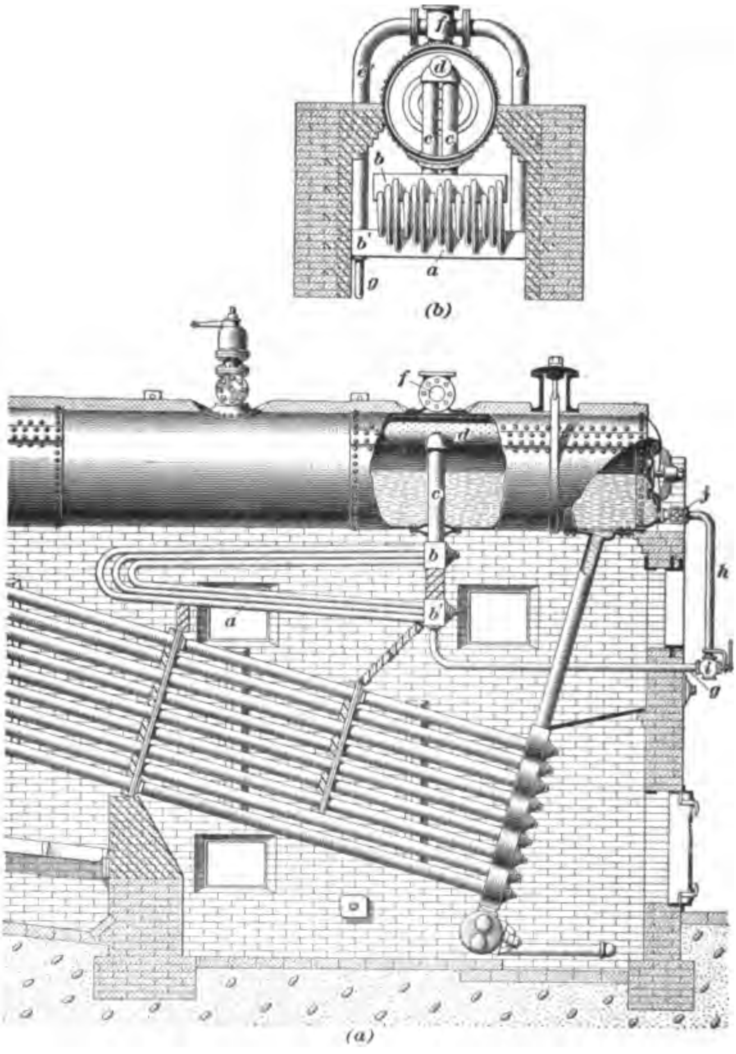


FIG. 23

valves *i* and *j* for the purpose of filling the superheater with water when not in use, as is the case when getting up steam or when the engine is not running. To put the superheater into service, the water is drained from it by means of the three-way valve *i*. Superheaters should be so constructed that all parts may expand and contract freely.

Superheaters are made either of wrought iron or of cast iron. The latter have the advantage that they can stand any temperature to which they are likely to be exposed, but they must be much thicker and heavier than wrought-iron superheaters. They require a higher temperature for the same useful effect, but at the same time they serve as reservoirs of heat and tend to equalize the temperature of the steam when the furnace temperature varies. The superheater should be accessible for removing the soot. The use of a jet of steam or air is the most effective means of cleaning the ribbed surface of a cast-iron superheater.

Generally, the requirements of a successful superheater are safety in operation, an economical use of the heat supplied, a minimum danger of overheating, and provision for free expansion with no exposure of the joints to the fire. Superheaters should be so located as to be cut out of service easily, repaired without interfering with the operation of the boiler, and applied readily to existing plants.

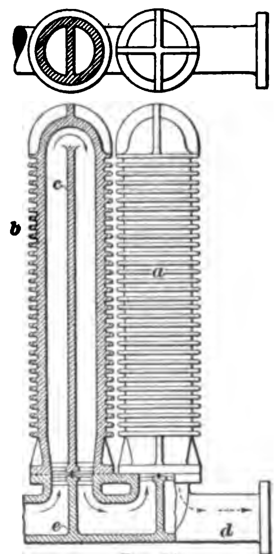


FIG. 24

40. In Fig. 24 is shown a portion of a cast-iron superheater that consists of a number of cast-iron cylinders *a*, having several exterior concentric ribs *b*. A partition wall *c* through the axis of the cylinder extends from the base to near the top, which is hemispherical in shape. The cylinders are set vertically, and are bolted to flanged openings in a horizontal cast-iron pipe *d* having partitions *e* that match those of the cylinders. The arrangement is such that the steam, in passing along the main pipe, is compelled to flow through all the cylinders in series, as shown by the arrows. The inlet and outlet of each cylinder being at the same end of the casting, perfectly free expansion and tight joints are secured.

FEEDING AND CLEANING APPARATUS

FEEDPIPING

41. The pipe system through which a boiler or boiler plant receives its water supply may be divided into two parts: the *external system* and the *internal system*. The external system comprises the piping required to take the feedwater from its source of supply and deliver it at the boiler. The internal system consists of the pipes leading from the outside of the boiler to the point of delivery.

42. The **external feed system** comprises the suction pipe of the feed-pump or injector and the delivery pipes or feedpipes that deliver and distribute the water to the different boilers. The suction pipes should be as short and free from bends as possible, especially when the water must be lifted some distance from the source of supply. It is generally very difficult to lift water to a greater height than 24 feet at sea level, unless the pump is in excellent condition; if the water must be lifted higher, it is usually better to locate the pump farther down, excavating for it if necessary. The suction pipes should also be perfectly air-tight. When the feedwater is taken from a city water supply, it usually comes to the pump or injector under some pressure; the feed-apparatus can then be located where most convenient.

43. The arrangement of the feedpipes naturally depends on the number of boilers to be supplied by the feed-apparatus and on the extent to which the pump or injector is required to supply water to any one or all of several boilers.

A good piping arrangement for a horizontal return-tubular boiler is shown in diagrammatic form in Fig. 25. The feed-pipe enters the front boiler head in the middle, directly above the top row of tubes. An elbow is placed at *a*, and

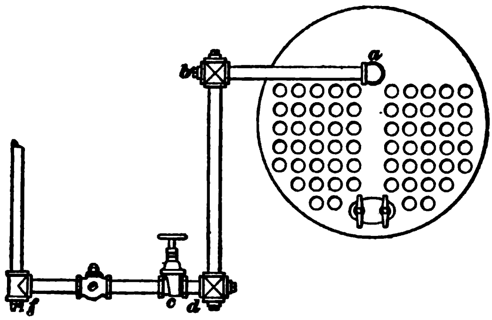


FIG. 25

the pipe is carried through the setting, a cross being placed at *b*. The pipe is now dropped sufficiently low to bring the stop-valve *c* into a convenient position. A cross is placed at *d*. A check-valve *e*, preferably of the horizontal

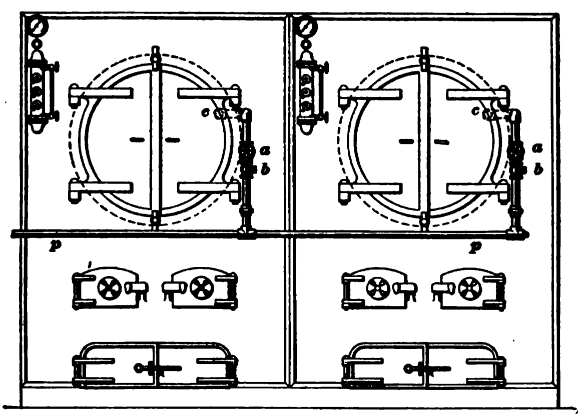


FIG. 26

swing-check pattern, prevents the return of the feedwater. Beyond the check-valve the pipe may be run as considered most convenient to reach the boiler feeder. The openings in the crosses *b* and *d*, and in the **T** *f*, opposite the pipes, are closed by brass plugs. On removing these plugs, the pipes can be cleared of sediment by running a rod through them.

Where feedpiping passes through the wall of the setting, or is exposed to the gases of combustion, extra heavy piping and fittings or brass piping and brass fittings should be used.

44. In Fig. 26 is shown an ordinary method of arranging the feedpipes where two boilers are supplied by the same boiler feeder. The main pipe *pp* running along the front of the boilers receives the feedwater discharged from the boiler feeder. Each boiler is supplied by a branch from the pipe *p*, entering the front head *c*. Each of these branches is provided with a stop-valve *a* and a check-valve *b*. The stop-valve shuts off the water from the boiler, while the check-valve allows the water to enter when the stop-valve is open, but prevents its return.

The stop-valve should always be placed nearest the boiler, thus allowing the check-valve to be examined and repaired without shutting down the boiler. With the arrangement of feedpiping shown, the feedwater can be delivered simultaneously to both boilers or to either boiler separately.

45. An arrangement of feedpiping for a plant having six boilers in two batteries and two independent feed-pumps is

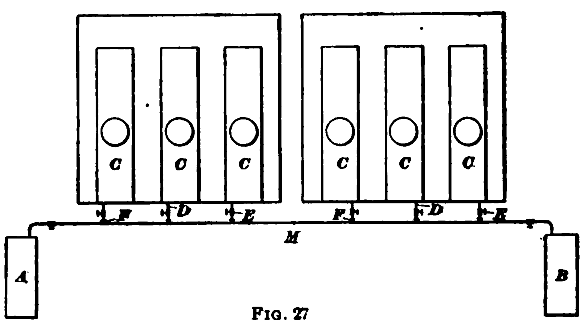


Fig. 27

shown in diagrammatic form in Fig. 27. Both of the feed-pumps shown at *A* and *B* are connected to the main feed-pipe *M*. This pipe has six branches, as *D*, *D*, one for each boiler. Each branch pipe has its own stop-valve *E* and check-valve *F*. There are two valves in the main feedpipe, one between each pump and the nearest branch pipe. With this arrangement, either or both pumps may be used for any or all boilers. Thus, if it is desired to use the pump *A* for all boilers, the valve in the main feedpipe near this pump is opened and the valve near the pump *B* is closed. The pump *A* will then deliver into any or all of the six boilers, depending on the manipulation of the stop-valves in the branch pipes.

46. When the feedwater goes through a feedwater heater before entering the boilers, it is usually advisable to provide by-pass connections, so that in case of accident to the heater it may be cut out of service without interference with the feeding. In many plants, the entire feed system is fitted in duplicate, in order to be prepared for emergencies. One system may then be supplied by injectors and the other by pumps.

47. The **internal feed system** is arranged in various ways. Its purpose is twofold: (1) to conduct the water to the proper point of discharge in the boiler; (2) to heat the relatively cold feedwater to nearly the temperature of the water in the boiler. As to the proper point of discharge, authorities differ considerably. Most engineers believe that the water should be discharged into the coolest part of the boiler and should be diffused by being delivered through a perforated pipe. Others discharge the feedwater into the steam space and use some suitable device to break the entering stream into spray.

When discharging into the water space of horizontal tubular boilers, a common and very satisfactory arrangement is to make the feedpipe enter the front head a little below the water-line and carry it to within a few inches of the rear head. The end of the pipe is closed and holes in the bottom of the pipe discharge the water downwards between the tubes. Or the water may enter through the bottom of the rear head; a horizontal pipe then carries it to within a few inches of the front head. It then passes through a vertical pipe up between the tubes to within a few inches of the water level, and returns to the rear through a horizontal pipe and is discharged downwards between the tubes. With this arrangement, the feedwater will be heated to the same temperature as the water in the boiler.

In plain cylindrical boilers, the feedwater usually enters the bottom of the front head. In good practice, it is then carried to the rear by a horizontal pipe and discharged upwards. In some plants, however, this is not done, and the water is discharged directly on the crown sheet. This is considered very poor practice by most engineers, since it subjects the plate, which is exposed to the most intense heat, to severe local stresses, which ultimately will strain the metal beyond its elastic limit and cause a rupture. In general, it is the common rule that feedwater should never be discharged on the parts of the boiler exposed to the most intense heat, nor should it be delivered in a solid stream against a plate; and, furthermore, it should be discharged in such a direction as to assist the circulation.

In flue boilers, the water may be discharged in the same manner as in return-tubular boilers. In vertical boilers, it is usually discharged into the water leg at the lowest point, although sometimes it is delivered about 2 feet above the crown sheet. In boilers of the locomotive type, it may be delivered into the lower portion of the cylindrical part or into the water legs below the grate.

FEEDING APPARATUS

48. Injectors.—On investigating the action of the **injector**, it will be found that dry steam at a given pressure enters the apparatus, passes through several contracted passages, and forces water past one or more check-valves into the boiler against a pressure equal to or greater than that which it had when beginning the operation. The steam, in forcing the water through the injector and into the boiler, gives up its heat and performs actual mechanical work as truly as though the steam acted on a piston and moved a pump plunger along with it. A current of any kind—be it steam, air, water, or other fluid—has a tendency to induce a movement, in the same direction, of any body with which it may be in contact. This mechanical principle underlies the action of the injector. The steam, moving with an extremely high velocity, imparts a portion of its velocity to the water and gives it sufficient kinetic energy to overcome a pressure even higher than the original pressure of the steam. The steam enters the injector at a high temperature with a great velocity, and when it strikes the cold water it is condensed, producing a partial vacuum. In striking the water at a high velocity, the steam gives up most of its kinetic energy to the water, thus carrying it forwards with a velocity that is still further increased by the water from the suction pipe rushing to fill the partial vacuum created by condensation of the steam. The velocity imparted to the water gives the latter sufficient energy to throw open the check-valves and enter the boiler against high pressure.

49. The action of an injector can best be explained by referring to Fig. 28, which represents a conventional form. In the figure, *A* is the injector; *B*, the boiler; *C*, the water supply; *D*, the steam pipe to the injector; *E*, the water-supply pipe; *F*, the delivery pipe, and *G*, the boiler check-valve that prevents the water in the boiler from flowing back into the injector. In the injector, *S* is the steam valve and *O* the overflow valve, from which a passage leads to the overflow outlet *x*.

To raise the water, the steam valve *S* is opened a small amount; this permits steam to flow out of the steam nozzle *a* into the tube *b*; then it passes through the overflow valve *O* and overflow *x* to the atmosphere. The steam flowing through the tube *b* and to the atmosphere carries with it some of the air from the water-supply pipe *E*, thus creating a partial vacuum there. As soon as sufficient vacuum is formed in the supply pipe, atmospheric pressure, acting on the water in the tank *C*, forces water into the injector, but as the water has not sufficient velocity to open the check-valve *G* against the pressure in the boiler, it passes through the overflow valve. The injector is then said to be primed.

When water appears at the overflow of the injector the steam valve is opened wide, which permits steam to pass through the injector in a larger volume. This steam, coming in contact with the stream of water that is flowing around the steam nozzle *a* into the tube *b*, forces the water with

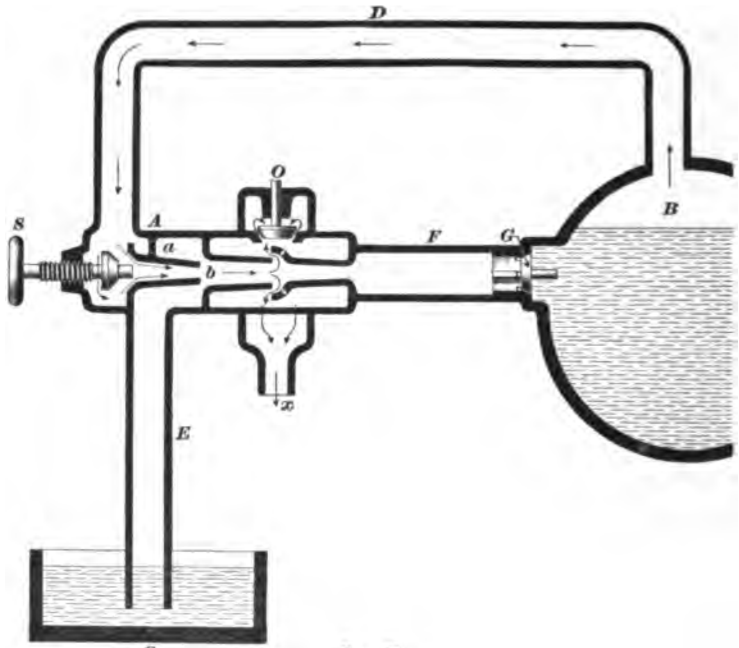


FIG. 28

increasing velocity through the tube *b*. Here, after imparting some of its kinetic energy to the feedwater, besides heating it, the steam is condensed, and in condensing forms a vacuum; owing to this vacuum, water is raised. The vacuum created varies with the pressure, being less with high pressures. Since there is a vacuum below the overflow valve *O*, that valve is held on its seat by atmospheric pressure, and as the steam and water discharge into a vacuum, they enter the injector at an increased velocity, which results in a final velocity of the water sufficient to carry it into the boiler.

In starting an injector, the water enters it at such a low velocity that the water is unable to open the boiler check-valve, so it passes out of the overflow valve to the atmosphere. The water continues to flow out of the overflow until the steam has imparted sufficient velocity to the inflowing water to enable it to force its way into the boiler.

50. Injectors may be divided into two general classes, *non-lifting* and *lifting injectors*. They differ from each other, as implied by the name, in that the one class is capable of lifting the water from a level lower than its own, which the other class cannot do.

Non-lifting injectors are intended for use where there is a head of water available; consequently, they must be placed below the water level of the supply tank, if one is used. When the water comes to a non-lifting injector under pressure, as from a city main, it can be placed in almost any convenient position close to the boiler. Non-lifting injectors resemble the lifting injector so much in their action that no description of them will be given.

Lifting injectors are of two distinct types, called *automatic* and *positive* injectors. Since positive injectors generally have two sets of tubes, they are also frequently called *double-tube* injectors.

Automatic injectors are so called from the fact that they will automatically start again in case the jet of water is broken by jarring or other means. They are simpler in construction than positive, or double-tube, injectors, and answer very well for a moderate temperature of feedwater supply and not too great a range in steam variation. They are very generally used on stationary boilers, and on portable boilers, such as are used to supply steam for traction engines.

In positive, or double-tube, injectors, one set of tubes is used for lifting the water, while the other set forces the water thus delivered to it into the boiler. They will lift water to a greater height than the automatic injector.

51. The construction of an automatic injector is shown in Fig. 29. Steam from the boiler enters the nipple *v*, passes into the nozzle *r*, and then into the conical combining tube *s*. In rushing past the annular opening between *r* and *s*, the steam creates a partial vacuum and causes water to flow through *p*, filling the space surrounding the lower end of *r* and the upper end of *s*. The nipple *p*, which is shown at the right-hand side, is really located in the rear. At first, the mingled steam and water do not flow to the boiler, because the water has not acquired sufficient velocity; but after the tube *y*, the space surrounding it, and the feed-delivery pipe attached to the nipple *z* are filled, the mingled steam and water force the check-valve *q* open and pass through the overflow *t*. As soon as the jet of water passing through the combining tube *s* has acquired sufficient velocity, the boiler check-valve is forced open and the water commences to enter the boiler. In consequence, no more water will pass through the space around the lower end of *s* and the upper end of *y*, and there being no pressure in this space, the overflow valve *q* will close. The overflow valve is kept closed by the atmospheric pressure, for while the injector is working steadily there will be a partial vacuum in the space around *s* and *y*.

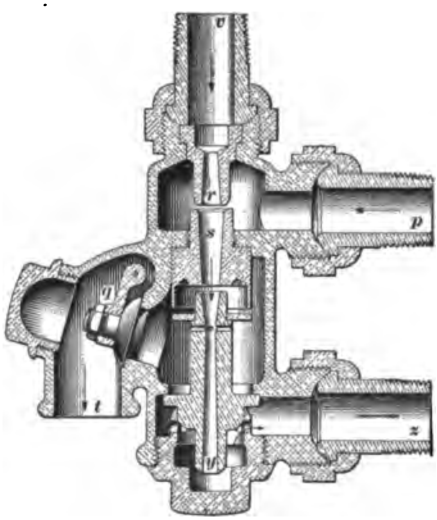


FIG. 29

To start the injector, all that is required is to turn on the steam and water. If the steam supply is too great, steam will issue from the overflow; if the water supply is too great, water will issue. Should the jet of water be broken, that is, fail to enter the boiler, the overflow valve will lift and the mingled water and steam will come out of the overflow until the jet has acquired sufficient force to enter the boiler again, when the overflow valve will close for the reasons given.

The automatically closing overflow valve is the distinguishing feature of the automatic injector, and in some form or other is found in all injectors of this class.

52. Steam Loop.—A certain apparatus, consisting chiefly of pipes and used for automatically returning the

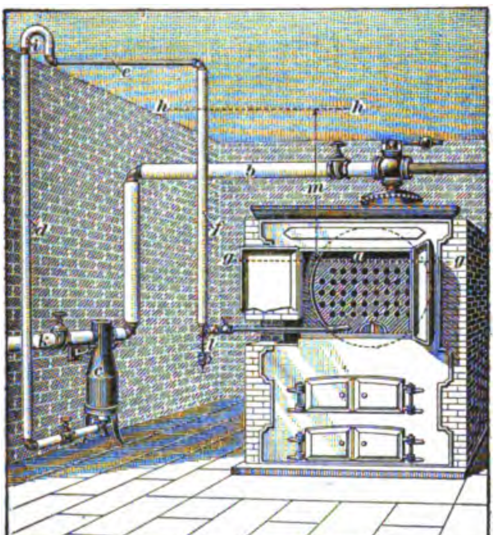


FIG. 30

water of condensation from a steam pipe, steam-heating system, steam separator, etc. to the boiler, is known as the **steam loop.** It is usually employed for lifting water from a point below the boiler water-line to such a height above it that the water will flow into the boiler by gravity. Its construction when applied to a separator is shown in Fig. 30, and if its principle of operation is understood, the loop can easily be modified to suit different conditions. The loop consists essentially of a riser *d*, a bend *i* acting as a check, a so-called horizontal *e*, a drop leg *f*, and a check-valve and globe or gate valve in the pipe connecting the drop leg to the boiler. The check-valve opens toward the boiler, and should never be omitted.

When in operation, the condensation of steam in the horizontal *e* reduces the pressure therein below that in the separator and thereby causes a current of mingled steam and water to flow from the separator *c* up the riser *d* into the horizontal, from which the water flows into the drop leg. For every pound difference between the pressure in the horizontal and that in the separator, a solid column of water would be lifted up the riser a distance of about 2.3 feet. Under working conditions, however, the contents of the riser *d*, consisting of a mixture of water particles and steam, weigh much less than water, and hence will be lifted to a much greater height. Thus it is evident that the action of this device depends on the condensation of steam in the horizontal and on the difference between the density of the mixture in the riser and that of the water in the drop leg. A continuous flow of mingled steam and water is insured by making the riser *d* less than the maximum height to which the contents of the riser could be lifted by a given pressure difference, the height of the drop leg *f* being made such as to give sufficient head to the water therein to force its way into the boiler.

The entrained water and water of condensation from the horizontal rises in the drop leg until the pressure due to its head, being added to the pressure in the horizontal, is sufficient to overcome the pressure within the boiler. The check-valve is then forced open and the water flows into the boiler until the water in the drop leg has dropped so low that the pressure exerted by it is insufficient to force it into the boiler. The check-valve then closes. The head of the water is measured between the water level *g g* of the boiler and the level at which the water stands in the drop leg. Thus, if the water stands at the level *h h*, the head is given by the distance *m*. This distance, from which the height of the drop leg can be determined, depends on the difference in pressure existing at the separator and the boiler pressure. In practice, about 2.5 feet should be allowed for each pound difference in pressure. As steam loops are liable to become air locked or flooded with water and thus rendered inoperative, their use is not recommended in places where there is not a skilled engineer in charge.

53. The **pumps** used for boiler feeding are usually direct-acting steam pumps, though in some cases power pumps are preferred. The former are independent, while the latter class are driven by belting from a line of shafting, and hence cannot be used except when the engine is running. The power pump is more economical than the steam pump, but the independence and convenience of the latter have brought it into general use.

54. The size of pump to feed a given boiler plant is easily determined by calculating the steam required per hour. This is done approximately by multiplying the rated horsepower of the boilers by 30, and adding, perhaps, 15 or 20 per cent. as a margin of safety. The pump should be large enough to feed the boiler under ordinary conditions by running continuously at slow speed, say thirty or forty strokes per minute.

Having given the feed-water consumption and the number of strokes per minute, the size of the water cylinder is readily found. For example, suppose that it is required to find the size of a pump to deliver 4,800 pounds of water per hour, the number of strokes under ordinary conditions being 30 per minute. There are approximately 62½ pounds of water in a cubic foot; hence, the water fed per minute in cubic feet is

$$\frac{4,800}{62\frac{1}{2} \times 60} = 1.28 \text{ cubic feet}$$

The water displaced per stroke must, therefore, be

$$\frac{1.28}{30} \text{ cubic feet} = \frac{1.28}{30} \times 1,728 = 73.73 \text{ cubic inches}$$

This would give a length of stroke of 6 inches and a cylinder diameter of 4 inches, nearly.

CLEANING DEVICES

55. Bottom Blow-Off.—For the double purpose of emptying the boiler when necessary and of discharging the loose mud and sediment that collect from the feedwater, each boiler is provided with a pipe that enters the boiler at its lowest point. This pipe, which is provided with a valve or cock, is commonly known as the **bottom blow-off.** The position of the blow-off pipe varies with the design of the boiler; in ordinary return-tubular boilers, it is usually led from the bottom of the rear end of the shell through the rear wall. Where boilers are supplied with a mud-drum, the blow-off is attached to the drum.

56. While in many boiler plants globe valves are used on the blow-off pipe, their use is objectionable, since, though tightly screwed down, the valve may be kept from closing properly by a chip of incrustation or similar matter getting between the valve and its seat. As a result, the water may leak out of the boiler unnoticed. Formerly, brass plug cocks were used almost entirely, but owing to their habit of sticking tightly they were superseded by globe valves and gate valves for high-pressure boilers.

Within the last few years plug cocks packed with asbestos have been placed on the market, the asbestos packing obviating the objectionable features of the plug cock. Many engineers now insist on the use of these cocks for the blow-off pipe. Gate valves are also used to some extent, but are open to the same objection as globe valves. In the best modern practice, the blow-off pipe is fitted with two shut-off devices. The one shut-off may be an asbestos-packed cock and the other some form of valve, or both may be cocks or valves, the idea underlying this practice being that leakage past the shut-off nearest the boiler will be arrested by the other.

57. The bottom blow-off pipe, when exposed to the gases of combustion, should always be protected by a sleeve made of pipe, or by being bricked in, or by a coil of plaited asbestos packing. If this precaution is neglected, the sediment and mud collecting in the pipe, in which there is no circulation, will rapidly become solid. Sometimes the blow-off pipe becomes so badly choked that on opening the blow-off cock the full steam pressure is not sufficient to clear the pipe.

The blow-off pipe should lead to some convenient place entirely removed from the boiler house and at a lower level than the boiler. Sometimes it may be connected to the nearest sewer; in many localities, however, ordinances prohibiting this are in force; the blow-off is then connected to a cooling tank, whence the water may be discharged into the sewer.

58. Surface Blow-Off.—As some of the impurities from the water in the boiler will float on the surface for a considerable time, a **surface blow-off** is often provided to remove them. In order to collect the scum, a shallow trough is sometimes placed in a boiler, with its top 2 or 3 inches below the surface of the water, when at its normal level, and a blow-off pipe connected to the bottom of the trough. But to be most effective, the skimming should be continuous.

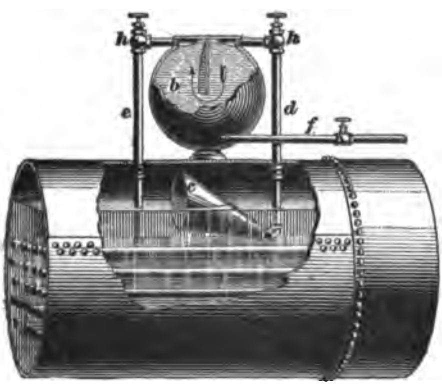


FIG. 31

An apparatus for this purpose is shown in Fig. 31. It consists of a globular vessel *b* placed above the boiler and containing a vertical diaphragm that divides it into two chambers. To the top of one chamber is attached a pipe *d* leading to a funnel *c* arranged at the surface of the water in the boiler to collect the scum. From the top of the other chamber a pipe *e* extends to a point near the bottom of the water space in the boiler. A blow-off pipe *f* is attached to the bottom of the vessel *b*.

When the valves are opened in the blow-off and skimmer pipes, the steam pressure forces some water with the surface scum into the globe. If the blow-off valve is now closed and the valve in the return pipe *e* opened, the water will return to the boiler, leaving the scum in the bottom of the globe. The circulation through the globe and the two vertical pipes is caused by the difference in density of the contents of the pipes. The water that enters the skimmer contains steam, and hence is lighter than that in the return pipe *e*. The water in the boiler is therefore in continual circulation through the globe. By regulating the flow so that the circulation is very slow, a considerable quantity of the impurities collected from the surface of the water in the boiler will accumulate in the bottom of the globe, and may be removed through the blow-off pipe with but little loss of water.

When magnesium salts are present in the feedwater, they will not settle, and it is necessary to place the globe upside down with the blow-off *f* on top. In this way, any accumulation of oil may also be discharged. Where both sinking and floating impurities are present, the globe is placed on its side with the diaphragm vertical, and a double blow-off used, one on top to remove the oil and the magnesium compounds, and the other at the bottom for lime and other sediment.

59. A good arrangement of the bottom blow-off for a return-tubular boiler is shown in Fig. 32. The blow-off pipe *a* has two right-angle bends of ample radius that render it springy. It is connected to the bottom of the boiler by a nipple screwed into the flange *b* and a right-and-left coupling. A pipe sleeve *c* protects the part of the pipe that would otherwise be exposed to the hot gases of combustion. An asbestos-packed plug cock *d* and angle cock *e* form shut-offs. The pipe *f*, leading to the sewer or blow-off tank, is connected to the angle cock *e* by nipples of suitable length and a flanged union *g*.

The usual diameters of blow-off pipes for tubular boilers are as follows: 1½-inch pipe for boilers up to 42 inches in diameter; 2-inch pipe for diameters up to 60 inches; and $2\frac{1}{2}$-inch pipe for larger power boilers.

60. Surface blow-off piping can advantageously be combined with the bottom blow-off piping in such a manner that a constant circulation may be maintained in the piping. The manner in which this is done is shown by the dotted lines in Fig. 32, where *h* is the surface blow-off pipe, connecting to the bottom blow-off pipe *a* beyond the cock *d*. When

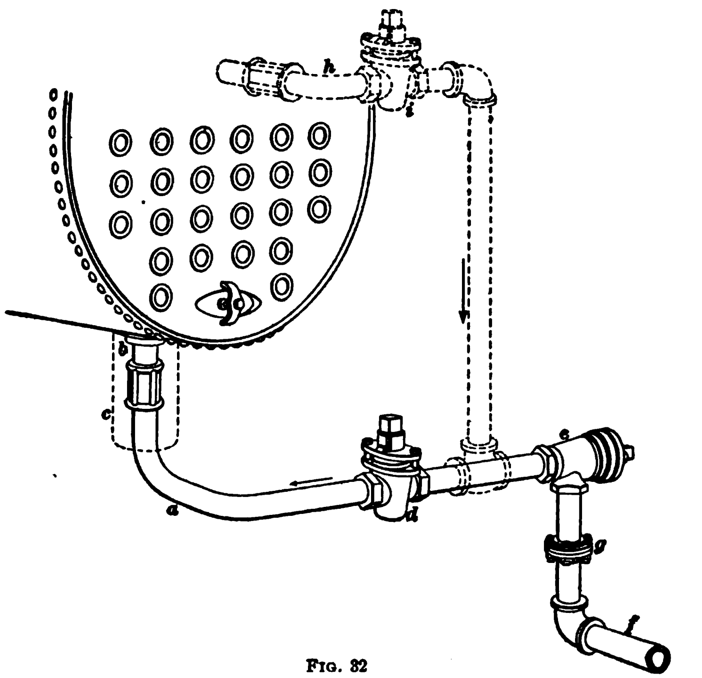


FIG. 32

neither blow-off is in use, the cocks *d* and *i* are open, but the blow-off cock *e* is tightly closed. The water then circulates in the direction shown by the arrows, preventing the accumulation of sediment in the blow-off pipe and subsequent rapid destruction of the latter by overheating. To use the surface blow-off, the cock *d* is closed and the cocks *i* and *e* are opened. To use the bottom blow-off, the cock *i* is closed and the cocks *d* and *e* are opened.

BOILER SETTINGS AND CHIMNEYS

BOILER SETTINGS

SETTINGS OF TUBULAR BOILERS

FOUNDATIONS

1. For all stationary boilers, unless the soil is of such a firm nature as to make a foundation unnecessary, it is customary to build brick, stone, or concrete foundations on which to erect the walls of the setting by which the boiler is enclosed and supported. Sometimes, the boiler is suspended from a structural-steel frame that rests on the foundation but is independent of the setting. The furnace and grate are arranged within the setting in such relation to the boiler as to secure economical results from the combustion of the fuel.

SUPPORTS FOR HORIZONTAL BOILERS

2. Boilers of the horizontal return-tubular type are supported by cast-iron brackets riveted to the shell and resting on iron plates embedded in the side walls, or by straps riveted to the shell and attached to overhead girders supported by the side walls.

3. When brackets are used, two are placed at each side. These brackets, in the best modern practice, are placed above the fire-line and protected by firebrick, it being inadvisable to expose them to fire, owing to the risk of burning them off. Rollers are generally placed between one pair of the brackets and the plates on which they rest, to provide for easy expansion and contraction. It is a general and good rule to set the boiler so that its expansion and contraction will not disturb the brick setting; in other words, the boiler should never be tied to the brickwork.

4. The method of supporting boilers by straps from overhead girders is employed for single boilers for the purpose of relieving the brickwork of the weight of the boiler and its contained water. The overhead girders are then supported on cast-iron or steel columns. This makes a somewhat expensive setting, which is claimed, however, to be less likely to crack than the setting receiving the weight of the boiler.

WALLS AND FIREBRICK LINING

5. In boiler settings, the walls have generally not only the weight of the boiler and its attachments to sustain, but they must also resist the varying stresses caused by the alternate heating and cooling of the entire masonry. For this reason, the foundations should be unusually heavy and the walls of ample thickness and properly lined with firebrick on the inside. Every sixth course of firebrick from the grate up should be a row of headers bonded into the masonry behind. By the term **headers** is meant that the bricks are set in with the ends exposed instead of the sides, as is the case with the other courses. This method enables the bricks between each row of headers to be renewed when necessary without having to tear down the entire wall.

BOILER FRONTS

6. There are two styles of settings for externally fired boilers; they are known as the **half-arch front** and the **full-arch, or full-flush, front** setting. In the half-arch setting, the smokebox projects beyond the boiler front.

The smokebox is made entirely of metal and is riveted or bolted to the boiler. In the full-flush front setting, shown in Fig. 1, the sides of the smokebox are formed, in part or wholly, by the brick setting and the end is enclosed by the boiler front, which contains doors for the smokebox, grate, and ash-pit.

The width of both styles of setting is the same, but the

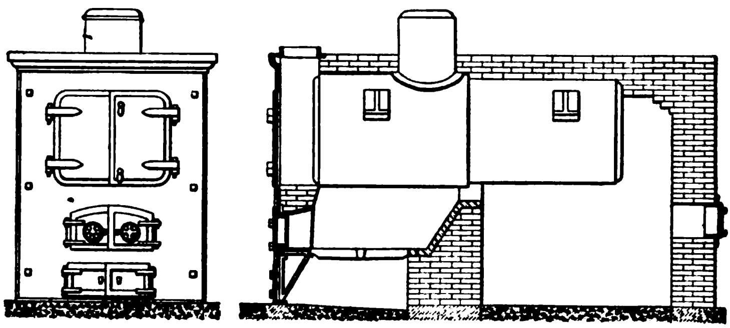


FIG. 1

half-arch setting is shorter and requires fewer bricks for its construction. The projection of the half-arch smokebox interferes to some extent with firing.

RETURN-TUBULAR BOILER SETTING

7. The setting of a 60-inch return-tubular boiler, as designed by the Hartford Boiler Insurance Company, is shown in Fig. 2. The foundation is made by digging down 3 or 4 feet and laying in heavy stonework; on top of this, the brickwork is laid. The side and rear walls are double, with a 2-inch dead-air space between the inner and outer parts. The inside wall *i* next to the furnace is faced with firebrick, as is also the bridge and all portions in direct contact with the flames.

The boiler is supported by cast-iron lugs *l* riveted to the shell. These lugs rest on iron plates *m* placed on top of the side walls. The front lugs rest directly on the plates, but the back lugs rest on rollers *o* of 1-inch round iron. The

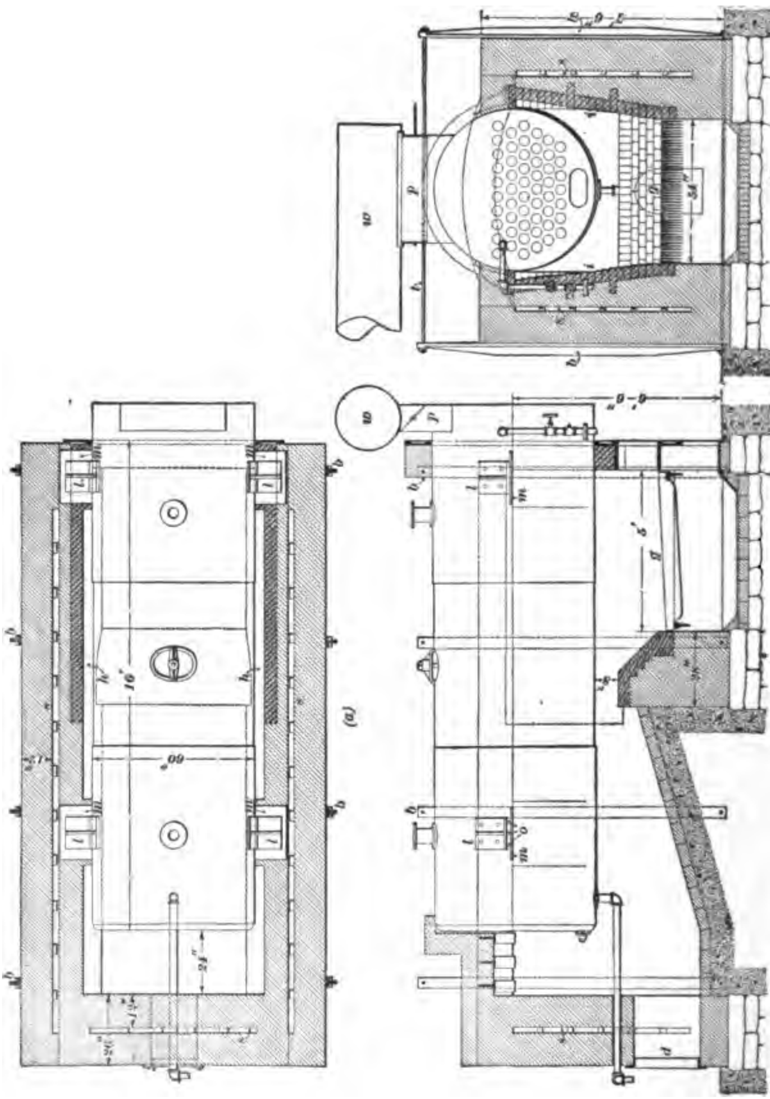


FIG. 2

boiler is thus free to expand and contract. The rear wall is 24 inches from the rear head of the boiler to allow the gases to enter the tubes. Above the tubes, however, the wall is built to meet the head and forms a roof for the chamber. The rear wall is provided with a door *d* for the removal of the dirt and soot that collect back of the bridge and to permit of inspection.

The grate *g* is placed 24 inches below the shell; this is a sufficient distance for anthracite, but for bituminous coal it may be 28 or 30 inches, and for wood as much as 36 inches, the grate being made narrower than for coal. The grate has a fall of 3 inches from front to rear, so that the fuel bed is thicker near the back end of the fire; this is believed to cause more even combustion, since the air has naturally a greater tendency to pass through the fire nearest the bridge, and on meeting a thick bed of fuel its passage is somewhat retarded.

The end of the boiler to which the blow-off pipe is attached should be set about 1 inch lower than the other end; this aids in the removal of mud and sediment and makes it possible to drain the boiler thoroughly.

The brickwork comes in contact with the shell at the level of the center of the upper row of tubes; this prevents the gases from coming in contact with the plates above the water-line. Sometimes a brickwork arch is constructed over the top of the boiler to allow the gases to pass back to the rear through the flue thus formed. This practice is risky, as it may lead to the overheating of the upper plates. The parts of the boiler not completely covered by water should never be exposed to the action of the fire or of the hot gases of combustion.

The brickwork is strengthened by buckstaves *b* held together by tie-rods *t*. The buckstaves are usually made of cast iron for small boilers and cheap work, but for large boilers and the better grade of work they are made of wrought-iron channel or angle irons. It will be noticed that in this case the flue pipe *p* is rectangular, but that the pipe *w* leading to the chimney is cylindrical.

Radiation of heat from the top of the boiler is prevented by a covering of brick, asbestos, or hair felt. The latter is generally laid in double sheets, so that the joints lap, and is easily removed for inspection of the boiler.

MISCELLANEOUS BOILER SETTINGS

8. The settings of the various types of water-tube boilers have been shown in the figures accompanying the descriptions given in *Types of Steam Boilers.* Internally fired boilers require a setting only for the purpose of forming a support and covering. Boilers of the Cornish and Lancashire type are so set that the boiler rests on two narrow ridges of fire-brick, which form the boundary between the bottom and side flues. The main point to be regarded in setting this type of boiler is to avoid having an excessive brickwork surface in contact with the boiler shell; the brickwork is liable to collect moisture and lead to external corrosion.

Plain cylindrical and flue boilers are set in about the same manner as the return-tubular boiler. Sometimes, however, when the shells are extremely long, two or even more bridge walls are placed beneath the shell to keep the heated gases in contact with the boiler.

Vertical and locomotive boilers, when self-contained, require no setting. The vertical boiler is supported by the cast-iron base that forms the ash-pit. Firebox boilers, when stationary, are supported on cast-iron saddles.

BOILER-SETTING FITTINGS

FURNACES

9. General Requirements.—To secure the highest efficiency in hand firing, the **furnace** should be made easy of access and the boiler room should be maintained at an endurable temperature. Provision should be made for an abundant supply of air under the grate as well as for a supply of heated air over the fire. Leakage of cold air into the combustion chamber of the furnace should be prevented.

Ample space should be provided between the grate and boiler to effect the complete combustion of the gases before

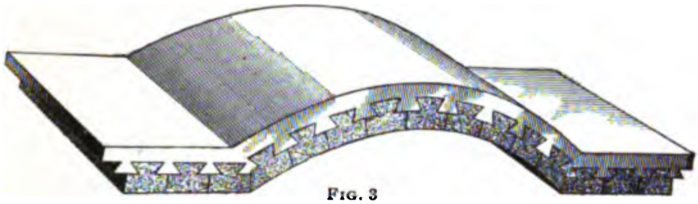


FIG. 3

they come in contact with the water-heating surface. Precautions should be taken to prevent loss of heat by radiation.

10. Furnace Mouth.—In order that the intense heat of the fire may not destroy the boiler front by warping and

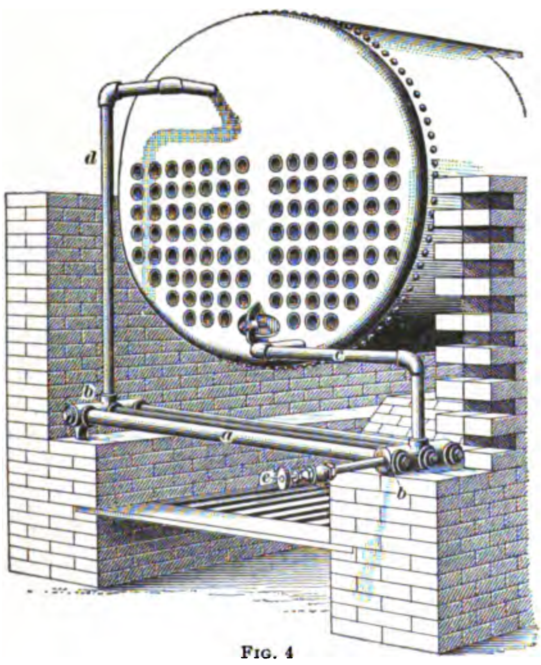


FIG. 4

cracking it, the front must be protected by a firebrick lining, which is supported on what is known as the **dead plate.** An arch is formed on a level with the top of the furnace door; this arch is generally supported on a cast-iron plate of suitable form, and known as the **arch plate.** Since the arch plate is exposed to the intense heat of the fire, it will soon burn out unless protected.

Fig. 3 shows a protected cast-iron arch plate having especially made firebricks dovetailed into grooves, as shown. These firebricks can be easily renewed when burned out.

11. Some engineers prefer to use a so-called **water arch** instead of a protected arch plate. There are a number of designs of this device on the market, one of which is shown in Fig. 4. It consists of three steel boiler tubes shown at *a*, expanded into headers *b*, *b*, and connected to the water space of the boiler by the pipe *c* and to the steam space by the pipe *d*. The tubes at *a* are set at an inclination, as shown, being higher at the end connected to the steam space, in order to allow the steam generated to escape readily. A blow-off *e* is fitted to the lower header for blowing out mud and sediment. The firebrick lining of the front and over the fire-door is carried on the tubes *a*.

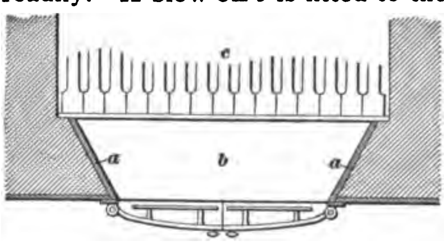


FIG. 5

12. It is sometimes considered desirable to protect the sides of the fire-door opening with cast-iron plates, called **cheek plates,** arranging them as shown at *a*, *a*, in the plan view of a furnace mouth in Fig. 5. In the illustration, the dead plate is shown at *b* and the grate at *c*. Some makers combine the dead plate, arch plate, and cheek plates into one casting. The objection to this is that the whole casting must be replaced when only one part, as, for instance, one of the cheek plates, is burned out.

13. The sides of the furnace mouth should taper from the fire-door opening to the sides of the furnace, as shown in Fig. 5. The ashes and clinkers can then be easily removed, since there is no place in the furnace mouth that cannot readily be reached.

THE BRIDGE WALL

14. The **bridge** is a low wall at the back end of the grate; it forms the rear end of the furnace, and causes the flame to come in close contact with the heating surface of the boiler. It is usually built of common brick and faced with firebrick, though in some cases it is made of wrought iron, with an interior water space communicating with the inside of the boiler. The passage between the bridge and boiler shell should not be too small; its area may be approximately one-sixth the area of the grate. The space between the grate and shell should be ample for complete combustion, and the distance between grate and boiler shell may be made about one-half the diameter of the shell.

GRATES

15. The **grate,** which is nearly always made of cast iron, furnishes a support for the fuel to be burned and must be provided with spaces for the admission of air. The spaces are distributed evenly all over the grate surface. The area of the solid portion of the grate is usually made nearly equal to the combined area of the air spaces; in other words, one-half the grate surface is air space and one-half serves to support the fuel. Grates are divided into two classes: *fixed grates* and *shaking grates.*

16. **Fixed grates,** as implied by their name, are stationary. The most common type of fixed grate is made of

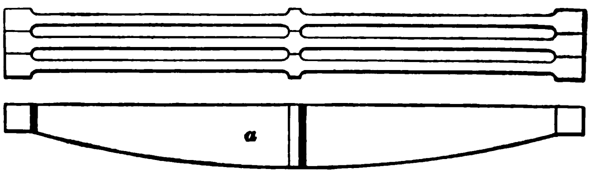


FIG. 6

single bars *a*, Fig. 6, placed side by side in the furnace. The thickness of the lugs cast on the bars determines the width of the open spaces of the grate. It is the general practice to make the thickness across the lugs twice the thickness of the top of the bar. Cast-iron bars are made about 2 inches deep at the ends and from 3 to 5 inches deep at the middle. For long furnaces, the bars are generally made in two lengths of about 3 feet each, with a bearing bar in the middle of the grate. Long grates are generally set with a downward slope toward the bridge wall of about $\frac{3}{4}$ inch per foot of length. This facilitates the admission of air to the rear of the grate; it also facilitates cleaning the grate.

Single grate bars are easily broken in transportation and handling; for this reason grate bars are often made as

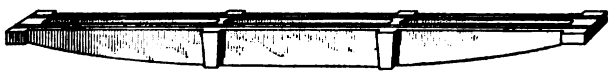


FIG. 7

shown in Fig. 7. Two bars are united in a single casting, which is not so fragile as a single bar.

17. The width of the air space, and hence the thickness of the grate bar, depends largely on the character of the fuel burned. For the larger sizes of anthracite and bituminous coal, the air space may be from $\frac{5}{8}$ to $\frac{3}{4}$ inch wide, and the grate bar may have the same width. For pea and nut coal, the air space may be from $\frac{3}{8}$ to $\frac{1}{2}$ inch, and for finely divided fuel, like buckwheat coal, rice coal, bird's-eye coal, culm, and slack, air spaces from $\frac{3}{16}$ to $\frac{3}{8}$ inch may be used. When these small air spaces are used, the grate bar, if made as shown in Figs. 6 and 7, must have the bars so thin in proportion to their length that they will warp and twist and a large number of the bars will soon break, especially when the rate of combustion is high.

18. To overcome this objectionable feature, the grate bar shown in Fig. 8, and known as the *herring-bone grate bar*, was designed, and in many places it has almost entirely superseded the ordinary grate bar. Owing to the angular shape of the cross-pieces of the bars, they are free to expand and contract. Being quite short and of small depth in comparison with the ordinary grate bar, there is very little danger of excessive warping of the bars; in consequence, they will usually far outlast a set of ordinary grate bars. Since there are only a few large bars for the grate, it is easier to replace a broken bar. Herring-bone grate bars can be obtained in a great variety of styles and with different widths of air

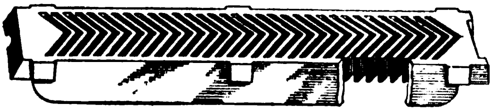


FIG. 8

spaces. They are usually supported on cross-bars, and, like many other forms of grate bars, may be arranged with trunnions, so as to rock the individual bars by means of hand levers, or to operate the bars in groups, so that several bars may be dumped as a unit.

19. A form of cast-iron grate bar especially adapted to burning sawdust is shown in Fig. 9. The bar is semicircular in cross-section and is provided with circular openings for the introduction of air. As in other types of grate bars, lugs are cast on each side of the bar to serve as distance pieces in providing air spaces between the bars.

Wrought-iron grate bars are rolled from a single bar; they have a head and web and are uniform in depth. They are

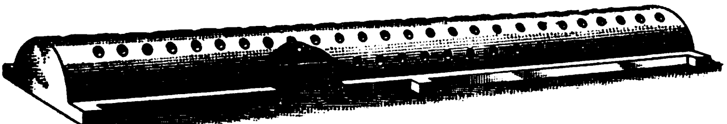


FIG. 9

made up in sets and riveted together, with distance pieces between the bars to form the required air space. Hollow grate bars through which water circulates are sometimes used.

20. In general, a grate bar should be especially suited for the kind of fuel to be burned. Thus, if very fine coal is to be burned, a grate bar having small air spaces and supports should be used, since otherwise a large percentage of the fuel will fall into the ash-pit. On the other hand, for the large sizes of coal, it is advisable to provide bars having large air spaces, using the largest air space when caking coals are to be burned. Some varieties of bituminous coal will *cake*, that is, fuse together to a considerable degree, and the ashes and clinkers formed will be of such size that a large part of them cannot pass through the air spaces unless these are large; the grate thus becomes clogged, shutting off the air from the fire. This reduces the rate of combustion and evaporation. When putting in grate bars, they should not be fitted in tightly, but plenty of room should be given to allow them to expand.

21. The front ends of the grate bars are usually supported on the *dead plate*, which is a flat cast-iron plate placed across the furnace just inside the boiler front and on a level with the bottom of the furnace door. The purpose of the dead plate is twofold: (1) It forms a support for the firebrick

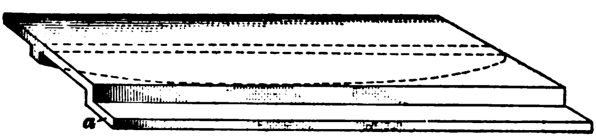


Fig. 10

lining of the boiler front; (2) it forms a resting place on which bituminous coal may be coked before it is placed on the fire. Experience has shown that most bituminous coals can be burned to the best advantage if they are coked first, by being exposed to the heat of the fire. To support the grate bars, the inner edge of the dead plate is either beveled or a lip is provided, as at *a*, Fig. 10. The dead plate should be at least 8 inches wider than the furnace, and be heavily ribbed to stiffen it. The supporting of the grate bars on a lip of the dead plate is objectionable, as ashes will soon get in between the ends of the bars and the dead plate and become hard, in consequence of which expansion of the grates will push the dead plate against the boiler front and in many cases break it. Too much care cannot be exercised to insert grate bars in such a manner that they can expand freely and without detriment to the boiler setting.

22. In the best modern practice, the grate bars are supported on *bearing bars*, made as shown in Fig. 11. The ends *a*, *a* are usually built into the side walls of the furnace, but a much better practice is to provide a cast-iron box *a*, Fig. 12,

FIG. 11

built into the side walls, on the bottom of which the end of the bearing bar *b* rests, as shown. This allows the bearing bar to expand and contract freely and permits ready renewal.

23. The greatest objection to stationary grate bars is that with them the furnace door must be kept open for a considerable length of time when the fire is being cleaned. Ashes, cinders, and clinkers will collect on the grate, shut off the air supply, and thus reduce the amount of steam generated. To restore the fire, it needs to be cleaned. Cleaning fires with a stationary grate not only severely taxes the fireman, owing to the excessive heat to which he is exposed, but the inrush of cold air chills the boiler plates, thus producing stresses that in the course of time will crack them. To overcome these objections, grates have been designed that allow the fire to be cleaned without opening the furnace door. This is usually done by giving each grate bar a rocking motion.

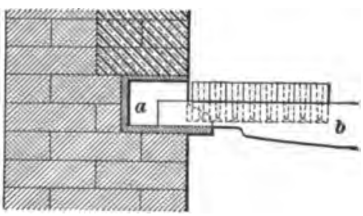


FIG. 12

24. Shaking Grates.—There are many designs of **shaking grates** for large steam boilers on the market, differing chiefly in detail and arrangement. A description of more than one is unnecessary, since the one described exhibits the characteristic features of most grates of this class.

Fig. 13 shows one form, in which the grate bars are hung on trunnions at each end and are connected together by

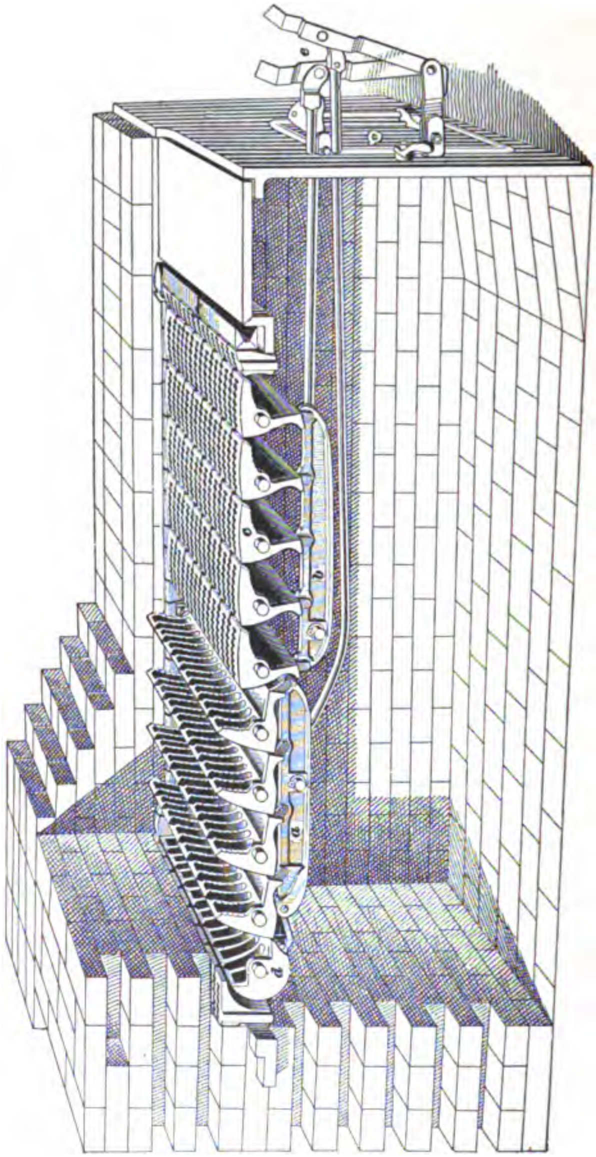


Fig. 18

bars *a* and *b*. Ordinarily they stand as shown in the right-hand half of the illustration. When it is desired merely to shake the fire and thus remove the bottom layer of ashes, the points *c* are moved from the level shown to the lowest position the connections will permit. The points follow the back of the bar immediately in front of them; thus no unusual opening is made through which fine fuel may fall into the ash-pit. The end bar *d* is curved to fit the frame. When the ashes have accumulated to a considerable thickness, or when they have fused together in a mass of clinkers, the points *c* are thrown upwards, as shown in the left-hand half of the illustration, thus forming a series of deep pockets that are closed at the bottom by the main rib, or back plate, of the grate bars. The act of throwing the points upwards breaks up the fused masses, which drop into the pockets and are discharged when the bars are returned to their normal position. The grate bars are operated by means of a handle fitting the levers shown at *e*. By means of these levers, either half of the grate can be operated independently. The two levers can, however, be locked together and all the grate bars worked back and forth simultaneously.

MECHANICAL STOKERS

25. Definitions and Classification.—A **mechanical stoker** is a power-driven rocking grate arranged so as to give a uniform feed of coal and to continuously rid itself of ashes and clinkers. The principal designs of mechanical stokers and automatic furnaces may be divided into two general classes, *overfeed* and *underfeed*. In the first class, the coal is slowly fed by some suitable mechanical device to a plate, where the volatile matter is driven off by the heat of the furnace and mixed with a suitable supply of air. The coal is then fed forwards on to grates, where it is burned. The mixture of gas and air is burned in a suitable combustion chamber, usually in as close proximity to the bed of burning fuel as is practicable. In the second class, the coal is forced by some mechanical device into a chamber under the

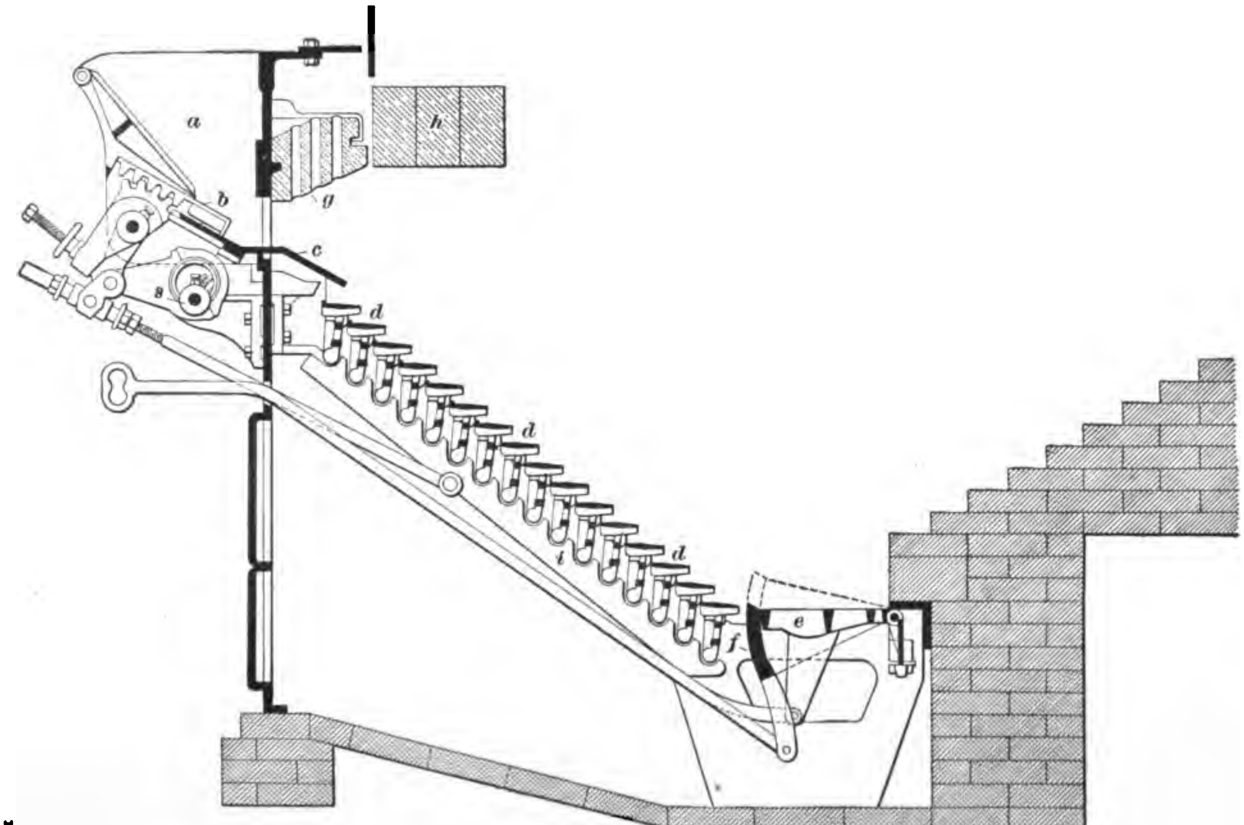


FIG. 14

mass of burning fuel in the furnace. The volatile matter is here driven off and mixed with a supply of air. The fuel is then pushed upwards by the fresh coal that is fed into the chamber and burns above the chamber and on suitable grates at the sides, on which it falls. The mixture of gas and air rises through the bed of burning coal above the chamber, and, being highly heated and thoroughly mixed, burns readily.

26. Overfeed Stokers.—In the earliest American **overfeed stokers**, the fixed carbon of coal is burned on inclined grates, consisting either of straight bars or a series of steps. The coal, after passing the dead plate, is pushed on to these grates, which are given a sufficiently rapid vibratory motion to feed it down at such a rate that practically all the carbon is burned before reaching the lower end, where the ashes and clinkers are discharged. In Fig. 14 is shown a sectional view of a stoker of this class. The coal is fed into the hopper *a*, from which it is pushed by the pusher plate *b* on to the dead plate *c*, where it is heated. From *c*, it passes to the grate *d d d*. This grate consists of cast-iron bars, which form a series of steps; each bar is supported at its ends by trunnions and is connected by an arm to a rocker bar *i*, which is slowly moved to and fro by an eccentric on the shaft *s*, so as to rock the grates back and forth; the grates thus gradually move the burning fuel downwards. The ashes and clinkers are discharged from the lower grate bar on to the dumping grate *e*, which can be lowered so as to drop them into the ash-pit below. A guard *f* may be raised, as shown by the dotted lines, so as to prevent coke or coal from falling from the grate bars into the ash-pit when the dumping grate is lowered. Air for burning the gases is admitted in small jets through holes in the **air tile** *g*, and the mixture of gas and air is burned in the hot chamber between the firebrick arch *h* and the bed of burning coke below.

27. Underfeed Stokers.—The stoker shown in Fig. 15 illustrates the principle of operation and the construction of the **underfeed stoker.** Coal is fed into the hopper *a*, from

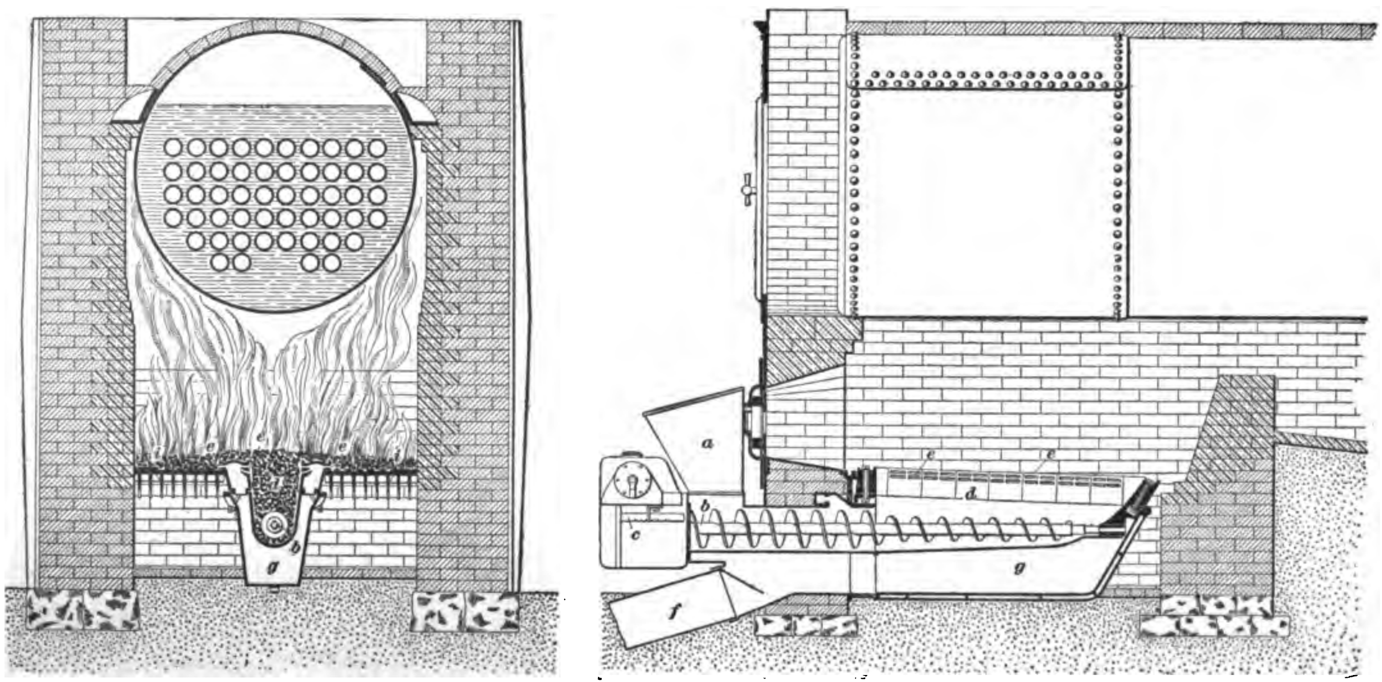


Fig. 15

which it is drawn by the spiral conveyer *b* and forced into the magazine *d*. The incoming supply of fresh coal forces the fuel upwards to the surface and over the sides of the magazine on the grates *i*, *i*, where it is burned. A blower forces air through a pipe *f* into the chamber *g* surrounding the magazine. From *g*, the air passes upwards through hollow cast-iron **tuyère blocks** and out through the openings, or **tuyères**, *e*, *e*, *e*. The gas formed in the magazine, mixed with the jets of air from the tuyères, rises through the burning fuel above, where it is subjected to a sufficiently high temperature to secure its combustion. Nearly all the air for burning the coal is supplied through the tuyères, only a very small portion of the supply coming through the grate.

The ashes and clinkers are gradually forced to the sides of the grate against the side walls of the furnace, from which they are removed from time to time through doors in the furnace front similar to the fire-doors of an ordinary furnace.

OIL-BURNING FURNACES

LIQUID-FUEL BURNERS

28. Liquid fuel can be burned under any boiler with very little alteration of the furnace, and in case of accident coal may be quickly substituted for the liquid fuel. Oil is, however, generally used only where the cost of producing power is greater with coal than with oil. On account of its high heating value, oil has been used to a limited extent on steam ships and has been tried on battle ships and locomotives. When injected by well-designed burners and burned in properly constructed furnaces, the combustion of crude petroleum is complete, giving off no smoke and no ash. In order that the combustion of the oil may be complete, it must be heated and atomized before coming in contact with the flame. The oil is, therefore, injected through a nozzle, or burner, into the furnace in the form of a fine spray mixed with steam or air. The proportions of oil and steam or air may be regulated so as to obtain complete combustion and freedom from smoke. The most common forms of burners use a jet of steam for spraying the oil and for inducing a current of air to mix with the spray. Air compressed to about 15 pounds pressure per square inch, or upwards, may be used instead of steam; but because of its greater convenience the latter is generally employed.

29. In the burner shown in Fig. 16 (*a*) and (*b*), the oil enters through the opening shown and passes through the inner nozzle *a*, where it comes in contact with the steam

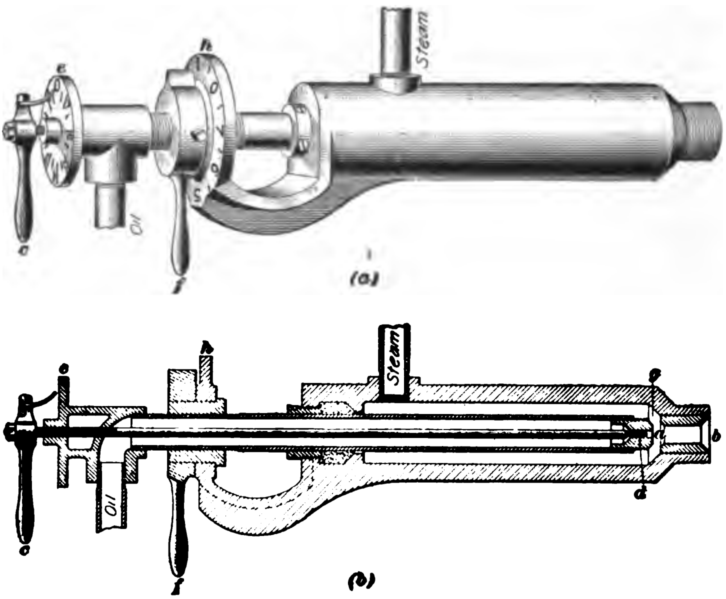


Fig. 16

jet that vaporizes the oil and projects it through the nozzle *b* into the furnace. The amount of oil passing through *a* is regulated by the handle *c*, which opens or closes the valve *d*, the disk and pointer at *e* indicating the amount of valve opening. Similarly, by means of the handle *f*, the steam passage *g* may be regulated, the amount of opening being indicated by the disk and pointer at *h*.

30. One method of installing an oil-burning system in a steam-boiler plant is shown in Fig. 17. Four ducts *a*, constructed of hollow tiles, are laid in the ash-pit, extending nearly to the bridge wall, and the ash-pit door openings are closed by brickwork *b* around the outer ends of the tiles. The forward bearer of the grate bars is dropped, about one-half of the forward set of bars removed, and a course of firebrick is laid with fireclay over the whole upper surface

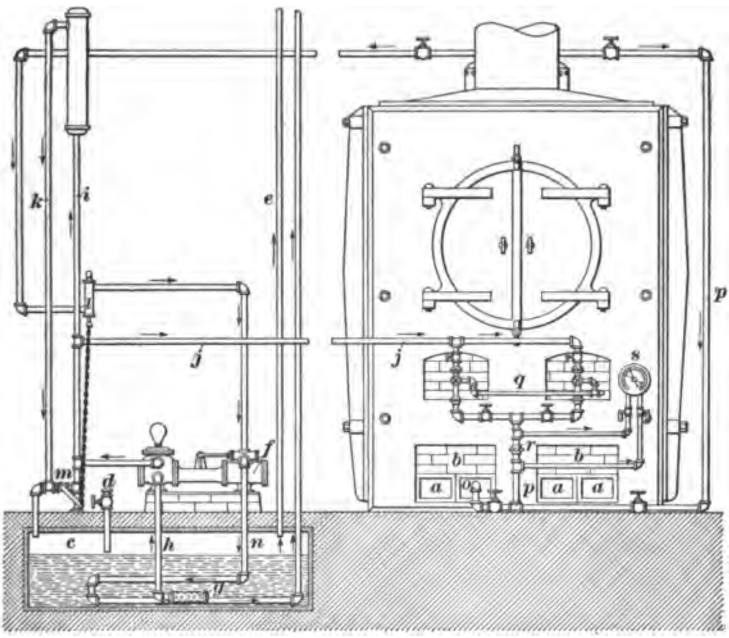


Fig. 17

of the grate. The air entering through the tiles passes under the grates to an opening in the front, and upwards into the combustion chamber. A checkerwork of about fifty loose firebricks is built in front of the bridge wall over the rear part of the grate, and on this the stream of burning oil spray strikes. The fire-doors are closed with brick, leaving a small opening for the insertion of the nozzles of the burners and another opening for lighting them.

The oil is stored in a closed tank *c* sunk in the ground. Oil is delivered from barrels into the tank through the pipe *d*; *e* is a vapor vent pipe. A steam pump *f* draws oil through the strainer *g* and pipe *h* and delivers it to the stand pipe *i*, from which it flows through the pipe *j* to the burners under a head of about 10 feet. The pump runs constantly, the surplus oil flowing back to the tanks *c* through the pipe *k*. Inserted in the steam pipe leading to the pump is a device *l* having a piston connected by a chain with a cock *m*, which is automatically opened to let the stand pipe be emptied of oil when the boiler is not under steam pressure. The exhaust pipe *n* passes through the tank to heat the oil and thus maintain it in a freely flowing fluid state in cold weather. A blow-off pipe *o* is used to drain water of condensation from the burner steam pipe *p* before starting the burner. To aid combustion, the burners used in this case are supplied with hot air at the point where the steam mingles with the oil. This air is supplied through the pipe *q* that is led through the brickwork in the fire-door into a brick flue over the grate bars, on one side, and down into the ash-pit. In the steam supply pipe to the burners, there is a union coupling *r* containing a perforated disk, through which the steam flows on its way to the burners, a duplex gauge *s* being provided for showing the pressure on each side of the orifice in the disk, thereby enabling the amount of steam supplied to the burners under a given pressure difference to be calculated.

CHIMNEYS

OPERATION, CONSTRUCTION, AND DESIGN

PRINCIPLE OF OPERATION

31. Draft.—It is well known that any volume of gas is lighter when heated than the same volume of gas when cool. Now, when the hot gases pass into the chimney, they have a temperature of from 400° to 600°, while the air outside the chimney has a temperature of from 40° to 90°. Roughly speaking, the air weighs twice as much, bulk for bulk, as the hot gases. Naturally, then, the pressure in the chimney is a little less than the pressure of the outside air. Consequently, the air will flow from the place of higher pressure to the place of lower pressure; that is, into the chimney through the furnace. The production of **draft** and the satisfactory operation of a chimney depends on this pressure difference.

32. Suppose, for example, that the average temperature of the gases in a chimney 150 feet high is 500° F. A pound of burned gases at 62° F. has a volume of 12.5 cubic feet; its volume at 500° is, then,

$$\frac{12.5 \times (500 + 460)}{62 + 460} = 23 \text{ cubic feet}$$

Therefore, a column of burned gases 1 foot square and 150 feet long will weigh 150 ÷ 23 = 6.52 pounds. A similar column of air at 62° F. will weigh 150 ÷ 13.14 = 11.42 pounds, nearly. Hence, the pressure of the draft is 11.42 − 6.52 = 4.9 pounds per square foot, or 4.9 ÷ 5.2 = .942 inch of water. It is evident that the pressure of the draft depends on the temperature of the furnace gases and the height of the chimney. The higher the chimney, the lower may be the temperature of the gases to produce the same draft, and the greater will be the economy of the furnace. In general, chimneys are not built much less than 100 feet in height.

33. Chimney draft is affected by so many varying conditions that no absolutely reliable rules can be given for proportioning chimneys to give a certain desired draft pressure, since the pressure required to force the air through the fire and to overcome the frictional resistances of the smoke flues and chimney to the passage of the gases cannot be determined beforehand with any degree of accuracy from purely theoretical considerations. For this reason, the rules given for chimney proportions are based on successful practice rather than on pure theory.

34. The draft produced by a chimney may vary from $\frac{1}{4}$ inch to 2 inches of water, depending on the temperature of the chimney gases and on the height of the chimney. Generally speaking, it is advantageous to use a high chimney and as low a chimney temperature as possible.

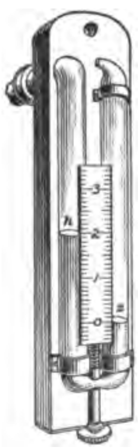


FIG. 18

35. The draft pressure required depends on the kind of fuel used. Wood requires but little draft, say $\frac{1}{2}$ inch of water or less; bituminous coal generally requires less draft than anthracite. To burn anthracite, slack, or culm, the draft pressure should be about $1\frac{1}{4}$ inches of water.

36. Measurement of Draft.—The intensity of the draft may be measured by means of a water gauge such as is shown in Fig. 18. As will be seen, it is a glass tube open at both ends, bent to the shape of the letter **U**; the left leg communicates with the chimney. The air outside the chimney, being heavier, presses on the surface of the water in the right leg and forces some of it up the left leg; the difference in the two water levels *h* and *z* in the legs represents the intensity of the draft and is expressed in inches of water.

CONSTRUCTION OF CHIMNEYS

37. Form.—The form or shape of a chimney has a pronounced effect on its capacity. A round chimney has a greater capacity for a given area than a square one, the frictional resistance to the flow of the waste gases of combustion in the latter being greater than in the former, while a narrow rectangular flue offers even greater frictional resistance to the passage of the gases. Calculations based on the requisite area of a round chimney should, therefore, be corrected to allow for the greater resistance offered by rectangular flues.

If the flue is tapering, the area for calculation is measured at its smallest section. The flue through which the gases pass from the boilers to the chimney should have an area equal to, or a little larger than, the area of the chimney. Abrupt turns in the flue or contractions of its area should be carefully avoided, as they greatly retard the flow of the gases. Where one chimney serves several boilers, the branch flue from each furnace to the main flue must be somewhat larger than its proportionate part of the area of the main flue. Loss of efficiency due to friction may be reduced to a minimum by making the inside of the chimney as smooth as possible.

38. Details of Construction.—Chimneys are usually built of brick, though concrete, iron, and steel are often used for those of moderate height. Brick chimneys are usually built with a flue having parallel sides and a taper on the outside of the chimney of from $\frac{1}{16}$ to $\frac{1}{4}$ inch per foot of height. The external diameter at the base of a brick chimney should be made about one-tenth of its height to insure stability. The thickness of the outer wall is usually one brick, or about 8 or 9 inches, for the first 25 feet from the top, increasing one-half brick for each additional 25 feet from the top downwards. If the inside diameter exceeds 5 feet, the top should be one and one-half bricks thick; if under 3 feet in diameter, it may be one-half brick in thickness for the first 10 feet from the top. A round chimney gives greater draft area for the same amount of material in its structure and exposes less surface to the wind than a square chimney. Large brick stacks are usually made with an inner core and an outer shell, with a space between them. The core is free to expand with the heat without distorting the shell. Sometimes the shell has iron rings laid up in the brickwork every 4 to 5 feet. Large brick chimneys are usually constructed with a series of internal pilasters, or vertical ribs, to give rigidity. The top of the chimney should be protected by a coping of stone or a cast-iron plate to prevent the destruction of the bricks by the weather; some ornamental finish is usually added at the top of the chimney.

39. Iron or steel stacks are made of plates varying from $\frac{1}{8}$ to $\frac{1}{2}$ inch thick. The larger stacks are made in sections, the plates being about $\frac{1}{4}$ inch thick at the top and increasing to $\frac{1}{2}$ inch at the bottom; they are lined with firebrick about 18 inches thick at the bottom and 4 inches at the top. Some designers prefer to use no lining on account of the likelihood of corrosion and the difficulty of inspection, and also because the inside of lined stacks cannot be painted.

On account of the great concentration of weight, the foundation for a chimney should be carefully designed. Good natural earth will support from 2,000 to 4,000 pounds per square foot. The footing beneath the chimney foundation should be made of large area. In compressible soils, piles should be used under the footing.

40. Brick Chimney.—In Fig. 19 is shown a brick chimney 162 feet high. The flue is 12 feet 3 inches in diameter at the base, tapers to 8 feet half way up, and remains of the same size to the top. The outer wall *a* is $17\frac{1}{2}$ inches thick for the first 50 feet, 13 inches for 60 feet, and 9 inches thick to the ornamental top *b*. The core *c* is $13\frac{1}{2}$ inches thick for 20 feet above the flue openings, 9 inches for the next 70 feet, and $4\frac{1}{2}$ inches for the remainder. There are two flue openings *d* and *e* with a deflecting partition *f* extending about two-thirds of their height between them.

41. Steel Chimney.—A steel chimney is shown in Fig. 20. It is 225 feet high above the foundation, 14 feet

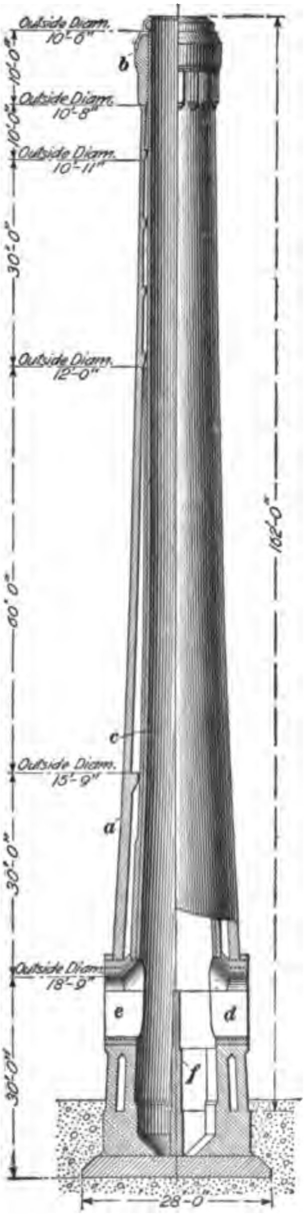

Outside Diam. 10'-6"
10'-0"
b
Outside Diam. 10'-8"
10'-0"
Outside Diam. 10'-11"
30'-0"
Outside Diam. 12'-0"
60'-0"
182'-0"
c
Outside Diam. 15'-9"
a
30'-0"
Outside Diam. 18'-9"
e
d
f
30'-0"
28'-0"

FIG. 19

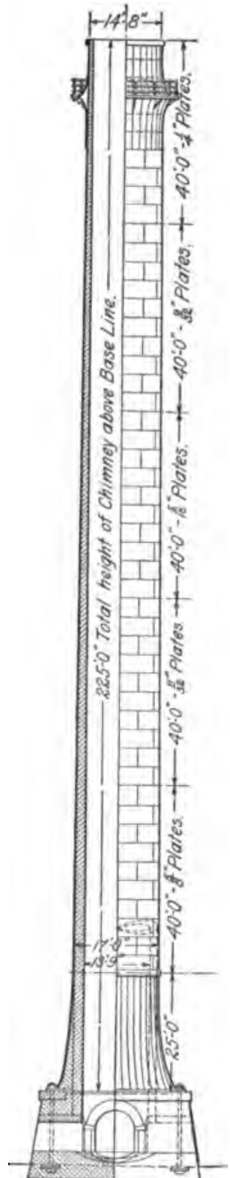

14'-8"
225'-0" Total height of Chimney above Base Line.
40'-0"-¼" Plates.
40'-0"-$\frac{9}{32}$" Plates.
40'-0"-$\frac{5}{16}$" Plates.
40'-0"-$\frac{11}{32}$" Plates.
40'-0"-⅜" Plates.
17'-0"
13'-9"
25'-0"

FIG. 20

8 inches inside diameter at the top, and is 13 feet 9 inches at the base. It is set on a foundation about 16 feet high, built of dimension stone laid in Portland-cement mortar. (*Dimension*, or *cut, stones* are stones that have been cut to dimensions in advance of laying.) The chimney is made of plates 4 feet high by about 6 feet long and $\frac{1}{4}$ inch thick for the first 40 feet from the top, increasing by $\frac{1}{32}$ inch per 40 feet for 160 feet. The first 25 feet at the bottom is made of $\frac{7}{16}$-inch plates, and tapers in a curved form from 17 feet diameter at the upper end to 27 feet diameter at the foundation bolt circle at the base. The chimney has a firebrick lining ranging in thickness from 18 inches at the bottom to $4\frac{1}{2}$ inches at the top. Several anchor bolts hold the chimney to the foundation and prevent it from blowing over. Four smoke flues, one on each side, enter the foundation near the bottom.

CHIMNEY CALCULATIONS

42. Height of Chimneys.—The relation between the height of the chimney and the pressure of the draft, in inches of water, is given by the following formula:

$$p = H\left(\frac{7.6}{T_a} - \frac{7.9}{T_c}\right)$$

in which p = draft pressure, in inches of water;
H = height of chimney, in feet;
T_a and T_c = absolute temperature of the outside air and of the chimney gases, respectively.

EXAMPLE.—What draft pressure will be produced by a chimney 120 feet high, the temperature of the chimney gases being 600° F., and of the external air 60° F.?

SOLUTION.—By the formula,

$$p = H\left(\frac{7.6}{T_a} - \frac{7.9}{T_c}\right) = 120\left(\frac{7.6}{460 + 60} - \frac{7.9}{460 + 600}\right) = .859 \text{ in. Ans.}$$

43. To find the height of chimney to give a specified draft pressure, the formula of Art. 42 may be transformed. Thus,

$$H = \frac{p}{\left(\frac{7.6}{T_a} - \frac{7.9}{T_c}\right)}$$

EXAMPLE.—Required, the height of the chimney to produce a draft of $1\frac{1}{8}$ inches of water, the temperature of the gases and of the external air being, respectively, 550° and 62°.

SOLUTION.—By the formula,

$$H = \frac{p}{\left(\frac{7.6}{T_a} - \frac{7.9}{T_c}\right)} = \frac{1.125}{\frac{7.6}{522} - \frac{7.9}{1{,}010}} = 167 \text{ ft. Ans.}$$

44. Area of Chimneys.—The height of the chimney being decided on, its cross-sectional area must be designed to carry off readily the products of combustion. The following formulas for finding the dimensions of chimneys are in common use:

Let H = height of chimney, in feet;
H. P. = horsepower of boiler or boilers;
A = actual area of chimney, in square feet;
E = effective area of chimney, in square feet;
S = side of square chimney, in inches;
d = diameter of round chimney, in inches.

Then,
$$E = \frac{.3 \text{ H. P.}}{\sqrt{H}} = A - .6\sqrt{A} \qquad (1)$$

$$\text{H. P.} = 3.33 E\sqrt{H} \qquad (2)$$

$$S = 12\sqrt{E} + 4 \qquad (3)$$

$$d = 13.54\sqrt{E} + 4 \qquad (4)$$

Table I has been computed from these formulas.

EXAMPLE 1.—What should be the diameter of a chimney 100 feet high that furnishes draft for a 600-horsepower boiler?

SOLUTION.—By formula 1,

$$E = \frac{.3 \text{ H. P.}}{\sqrt{H}} = \frac{.3 \times 600}{\sqrt{100}} = 18$$

Now, using formula 4,

$$d = 13.54\sqrt{18} + 4 = 61.44 \text{ in. Ans.}$$

EXAMPLE 2.—For what horsepower of boilers will a chimney 64 inches square and 125 feet high furnish draft?

SOLUTION.—By simply referring to Table I, the H. P. is found to be 934. Ans.

45. Maximum Combustion Rate.—The maximum rates of combustion attainable under natural draft are given

TABLE I

SIZE OF CHIMNEYS AND HORSEPOWER OF BOILERS

Height of Chimney, in Feet											Effective Area Square Feet	Actual Area Square Feet	Side of Square Inches	Diameter Inches
50	60	70	80	90	100	110	125	150	175	200				
Commercial Horsepower														
23	25	27									.97	1.77	16	18
35	38	41									1.47	2.41	19	21
49	54	58	62								2.08	3.14	22	24
65	72	78	83								2.78	3.98	24	27
84	92	100	107	113							3.58	4.91	27	30
	115	125	133	141							4.47	5.94	30	33
	141	152	163	173	182						5.47	7.07	32	36
		183	196	208	219						6.57	8.30	35	39
		216	231	245	258	271					7.76	9.62	38	42
			311	330	348	365	389				10.44	12.57	43	48
			402	427	449	472	503	551			13.51	15.90	48	54
			505	539	565	593	632	692	748		16.98	19.64	54	60
				658	694	728	776	849	918	981	20.83	23.76	59	66
				792	835	876	934	1,023	1,105	1,181	25.08	28.27	64	72
					995	1,038	1,107	1,212	1,310	1,400	29.73	33.18	70	78
					1,163	1,214	1,294	1,418	1,531	1,637	34.76	38.48	75	84
					1,344	1,415	1,496	1,639	1,770	1,893	40.19	44.18	80	90
					1,537	1,616	1,720	1,876	2,027	2,167	46.01	50.27	86	96

by the following formulas, which have been deduced from the experiments of Isherwood:

Let F = weight, in pounds, of coal per hour per square foot of grate area;

H = height, in feet, of chimney or stack.

Then, for anthracite burned under the most favorable conditions,

$$F = 2\sqrt{H} - 1 \qquad (1)$$

and under ordinary conditions,

$$F = 1.5\sqrt{H} - 1 \qquad (2)$$

For best semianthracite and bituminous coals,

$$F = 2.25\sqrt{H} \qquad (3)$$

and for less valuable soft coals,

$$F = 3\sqrt{H} \qquad (4)$$

The maximum weight of combustion is thus fixed by the height of the chimney; the minimum rate may be anything less.

EXAMPLE.—Under ordinary conditions, what is the maximum rate of combustion of anthracite coal if the chimney is 120 feet high?

SOLUTION.—By formula 2,

$F = 1.5\sqrt{120} - 1 = 15.4$ lb. per sq. ft. per hr. Ans.

EXAMPLES FOR PRACTICE

1. What should be the height of a chimney to give a draft pressure of $\frac{5}{8}$ inch of water, the temperature of the air being 60° F. and of the gases 440° F.? Ans. 107 ft.

2. A chimney is 135 feet high and 5 feet square inside; calculate the horsepower for which it will furnish draft. Ans. 851 H. P.

3. What is the maximum rate of combustion of best bituminous coal in a marine boiler with chimney stack 100 feet high? Ans. 22.5 lb.

4. Calculate the side of a square chimney 150 feet high that furnishes draft for boilers of 1,000 H. P. Ans. 63.4 in.

5. What draft pressure will a chimney 80 feet high furnish, the temperatures of the air and gases being, respectively, 60° and 600° F.? Ans. .57 in.

6. Under the most favorable conditions, what height of chimney will allow a maximum rate of combustion of anthracite coal of 23 pounds per square foot of grate per hour? Ans. 144 ft.

CHIMNEY FITTINGS

SMOKE-PIPE CONNECTIONS

46. The gases of combustion are conveyed from the boiler to the chimney either by a **smoke pipe,** generally made of sheet iron, or by a brick flue. For most high-pressure boilers, the smoke pipe is a simple round pipe, which should have an area at least 15 per cent. larger than the combined area of the boiler tubes. When boilers are set in a battery and connected to the same chimney, the different boilers are first united to a common duct *a*, Fig. 21, by

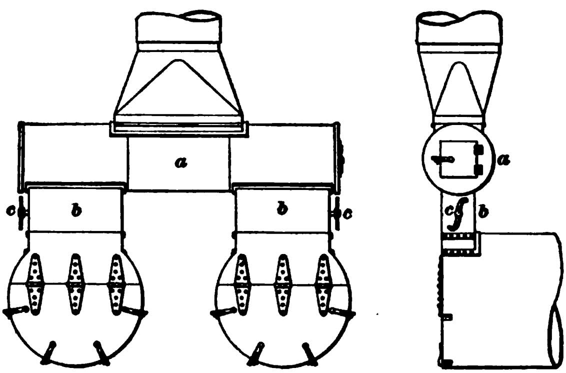


Fig. 21

branches *b*, *b*; the duct *a* and branches *b*, *b* are frequently called the **breeching.** Each branch must be provided with its own damper; the handles of the dampers are shown at *c*. The chimney is either placed directly on top of the duct *a*, as shown in the illustration, or a smoke pipe is led from the breeching to the chimney. Breechings are made in many forms to suit the existing conditions; the illustration given will serve as a suggestion. The area of the duct should equal the combined area of the branches; and the combined area of the branches should be about 15 per cent. larger than the combined area of the boiler tubes.

47. Smoke pipes, as well as exposed parts of the boiler, should always be covered with some good non-conducting material to prevent loss of heat. Boiler and smoke-pipe coverings are similar to the regulation pipe coverings, and are made in block shape from 1 to 3 inches thick. They are applied to the exposed shell of the boiler or to smoke flues by securing them with wire and then plastering over them to make a smooth, air-tight finish. Plastic asbestos is also used. A thin coat of the latter is placed on the shell of the boiler or smoke pipe, and then ordinary coarse-wire mesh is bound around and fastened to bond the material, after which a hard thick coat is plastered on and troweled smooth.

Some coverings are put on over an air space, this method not only insuring the most perfect insulation, but also preventing the impurities sometimes found in the covering composition from rusting the boiler. Metal lath is placed against the shell, or wire lathing may be used, with spacing nipples or bars. On this lathing, the blocks, or plastic covering material, are placed and secured and an outer coat is then applied.

48. Loss of heat from cylindrical boilers may be prevented by arching them over with brickwork, with an air space of 1 inch between the shell and the enclosing brickwork. A more convenient and practically as effective way is to cover the boiler with dry loam to a depth of from 4 to 6 inches. This covering may readily be removed for either boiler inspection or repairs.

DAMPER REGULATORS

49. In many steam-power plants, it is desirable that the steam pressure be kept practically uniform. For this purpose there have been designed numerous devices known as **damper regulators** which, operating on a change of the steam pressure in the boiler, automatically control the position of the smoke-pipe damper and thus regulate the volume of gases passing into the chimney. This, in turn, regulates the intensity of the fire and the generation of steam.

Damper regulators may be divided into four general classes:

1. Steam-actuated regulators in which the motion of a diaphragm under variation of steam pressure is transmitted, either directly or through some multiplying device, to the damper.

2. Steam-actuated regulators in which a piston is subjected directly to the boiler pressure, and, moving under a variation of pressure, turns the damper by means of suitable connections.

3. Steam-actuated regulators in which the steam in acting on a diaphragm causes a displacement of a valve, which admits steam into a cylinder, the piston of which is connected to a damper.

4. Hydraulically operated regulators in which the movement of a diaphragm under variation of the steam pressure operates an admission valve, admitting water under pressure to a cylinder, the piston of which is connected to the damper.

50. Damper regulators of the first class are relatively simple and inexpensive and well adapted for low-pressure heating work, giving a regulation close enough for the purpose.

The second class of regulator is cheap and simple; it is adapted for high-pressure work but will not give a very close regulation, owing to the fact that any variation in steam pressure sufficient to operate the device will cause the piston to move its whole length of stroke. This, in turn, causes the dampers to be either wide open or completely closed.

Regulators of the third class will regulate very closely, the makers of some such regulators guaranteeing that the motion of the damper from one direction to the other will change with a variation of steam pressure of $\frac{1}{4}$ pound per square inch, either way, from the point at which it is set to operate.

Regulators of the fourth class will also regulate very closely. Being dependent on water under pressure for their action, their application is limited to places where they can be connected either to a city water service or to a tank sufficiently high above the regulator to give the required pressure. They are sometimes connected directly to the water space of a boiler; while the damper regulator will operate when so connected, this method is open to the objection that it results in a waste of heat that may be quite large.

AUXILIARY APPLIANCES

MECHANICAL-DRAFT APPARATUS

PURPOSE AND APPLICATION

51. When the amount of coal to be burned per square foot of grate per hour exceeds the combustion rate under natural, or chimney, draft, it becomes necessary to resort to **artificial draft,** which may be produced by means of fan blowers or steam jets. According to the manner in which it is applied, it is known as *forced draft* or *induced draft.* In a forced-draft installation, the air is forced into the ash-pit by suitable means; in an induced-draft installation, a partial vacuum is created in the chimney. Both of these systems are in common use.

There are two methods of applying the forced-draft system, known, respectively, as the *closed stoke-hole* and *closed ash-pit methods.* The former is used in marine practice, while the latter is used for both stationary and marine service. The induced-draft method is also used for controlling the draft in both marine- and land-boiler furnaces.

52. The **closed stoke-hole method** consists in making the stoke hole of the vessel air-tight and forcing air into it by means of a blower until the stoke-hole pressure exceeds that of the atmosphere by the desired amount. Access to the stoke hole must be through air locks. The principal objection to this system is the great inrush of cold air into the furnaces when their doors are opened, resulting in the cooling of the furnace and starting leaks in the boiler.

53. The **closed ash-pit method** consists in providing air-tight ash-pits and forcing the hot air into the ash-pit by means of a blower. It is open to the objection that there is much resistance to the passage of air through the fuel, and that the flame may be forced out of the fire-doors. Another objection is that the fire-doors cannot be opened until the draft is shut off.

54. Induced draft is produced by an exhaust fan in the uptake or chimney. It is particularly adapted for use in conjunction with an **economizer,** a device for heating the boiler feedwater by means of the waste chimney gases. In some cases, the use of an economizer under natural draft reduces the draft pressure and the temperature of the flue gases to such an extent as to decrease the efficiency of the boiler because of incomplete combustion of the fuel. By the use of induced draft, the temperature of the flue gases may be reduced to any degree without affecting the draft pressure. When used in connection with an economizer, the gases of combustion are drawn from the furnace into the main flue, through the fan, and forced out of the stack. In this system, the stack is required to rise only a short distance above the roof. The economizer is usually placed in the main flue, between the furnace and the fan.

FORCED-DRAFT APPARATUS

55. Fan Blower.—For forcing air under pressure into the ash-pit, either a **fan blower** or steam jet may be used. While the installation of a fan blower is more costly than that of a steam jet, it is much more economical than the latter. A common construction of a fan blower is shown in Figs. 22 and 23. The shell or housing is made of steel plate, with a substantial base of cast iron or wrought iron. An outlet is placed at the desired point of the circumference, whence the air is discharged into the duct leading to the ash-pit. In the fan shown in Fig. 22 there is one inlet, which has the fan shaft in its center and is on the side farthest from the pulley and support. The fan shaft is supported in two bearings and carries the fan wheel within the casing. The usual construction of the fan wheel is shown in Fig. 23. Arms made of

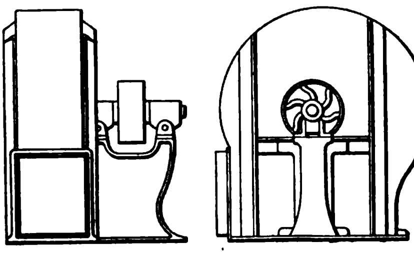


FIG. 22

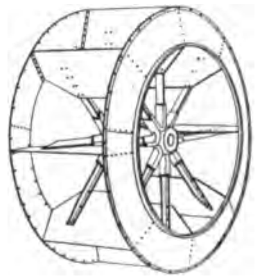


FIG. 23

T iron are fastened to the hubs and carry at their ends the blades. These blades are tied together by the side plates.

56. Ash-Pit Fixtures.—The air discharged by the fan may be introduced into the ash-pit through an opening in the bottom of the bridge wall, which is then built hollow, as shown in Fig. 24. The opening is closed by the

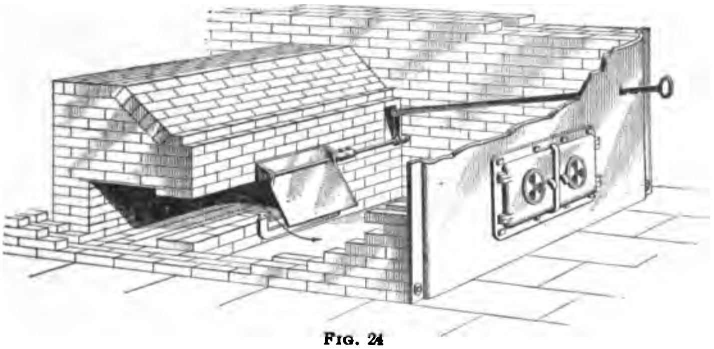


FIG. 24

damper shown. This arrangement is recommended for a new boiler plant. When the fan draft is applied to an old plant, the air may be introduced in front through an opening in the bottom of the ash-pit, as shown in Fig. 25. When the damper is closed, the ashes may readily be raked over it. The damper, when opened, serves to thoroughly distribute the air in the ash-pit.

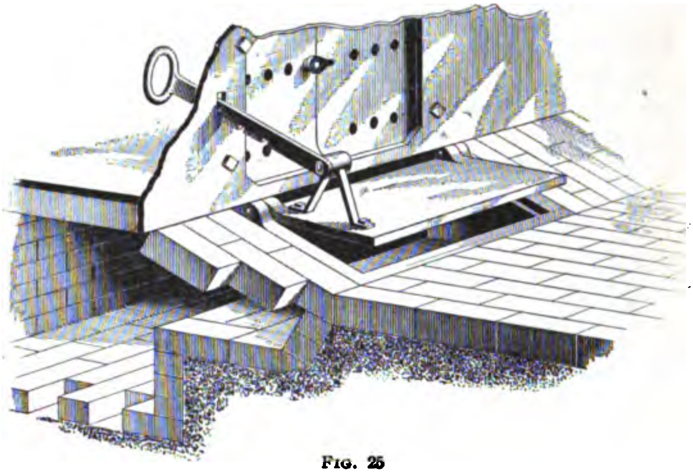


Fig. 25

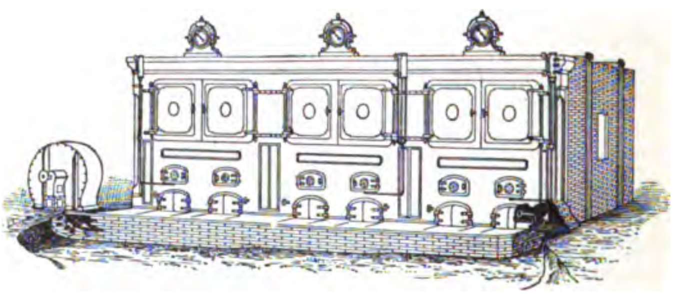


Fig. 26

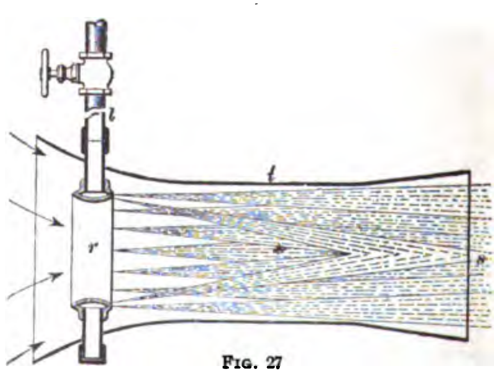


Fig. 27

57. Air Ducts.—The ducts leading to the ash-pit may be underground, as shown in Fig. 26. There is one duct along the front of the boilers, with one branch for each ash-pit. When a low first cost is essential, galvanized-iron pipes may be used for ducts, with the main pipe overhead and a branch pipe extending down to each boiler.

58. Steam Jet Blower.—An **Argand steam blower** for forcing air into the ash-pit is shown in Fig. 27. The blower consists of a long air tube *t* discharging from the end *s* below the grate. In the other end of the tube is placed a ring-shaped tube *r*, perforated on the right with small holes. Steam from the boiler is led into the ring by the pipe *l* and escapes in jets through the perforations, carrying air along with it into the ash-pit. This method of producing a forced draft is frequently used for burning the finer grades of anthracite and for bituminous slack.

INDUCED-DRAFT APPARATUS

59. A typical induced-draft installation, in which the fan *a* is connected directly to the engine *b*, is shown in Fig. 28.

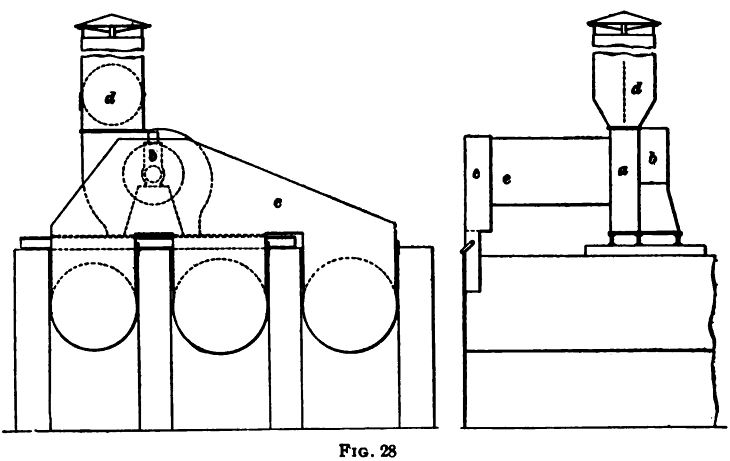


FIG. 28

The fan, when running, creates a partial vacuum in the uptake *c* and furnace, thereby causing the outside air, under the influence of its greater pressure, to flow into the ash-pit. The fan may be placed directly over the boilers, as shown in Fig. 28, discharging the gases into a short steel stack *d*; or it may be placed where most convenient. When applying induced, or suction, draft to an old plant using natural draft, the fan may be placed in connection with the smoke flue *e* in such a manner as to draw the gases from it and return them to it again at a point farther on, whence the gases may pass to the chimney. Suitable dampers serve to switch off the fan and pass the gases directly through the flue when the fan is not in service.

A partial vacuum may also be created in the chimney by means of a steam jet; this is a method commonly adopted in locomotives, in which the exhaust nozzle is placed in the stack, and the exhaust steam, by carrying with it the air in the stack, creates the draft. This method is seldom used in stationary practice.

ADVANTAGES OF MECHANICAL DRAFT

60. Mechanical draft possesses some advantages over natural draft that in many cases make it advisable to substitute it for natural draft. These advantages are as follows:

61. Adaptability.—Blowers may be applied under almost all circumstances and are independent of location and climatic conditions. They are used either to furnish draft independently of the chimney action, or to assist a chimney that does not create sufficient draft.

62. Ease of Control and Flexibility.—A fan may be automatically controlled to produce a very close approach to a uniform steam pressure by attaching an automatic damper regulator to the throttle valve of the engine. With natural draft, the intensity of the draft depends on the intensity of the fire, and is therefore least when the fire is low. With mechanical draft, however, the draft is independent of the condition of the fire, and, consequently, banked fires may be started up quickly. Furthermore, it can be readily adjusted for the combustion of different kinds of fuel and for widely varying combustion rates. Owing to the intensity of draft that may be created, not only can the low grades of coal be burned successfully, but the amount of steam generated by each boiler may also be greatly increased.

POWER REQUIRED

63. The horsepower necessary to furnish a forced draft may be calculated by the following formula:

Let p = pressure of draft, in pounds per square foot;
W = weight of fuel, in pounds per square foot of grate, burned per minute;
V = volume of air, in cubic feet per pound of fuel;
y = efficiency of draft apparatus.

Then,
$$\text{H. P.} = \frac{p\,W\,V}{33{,}000\,y}$$

EXAMPLE.—What horsepower is required to supply air at a pressure of 2½ inches of water to a total grate area of 120 square feet burning 20 pounds of coal per square foot per hour and requiring 220 cubic feet of air per pound of coal? Assume the efficiency to be 60 per cent.

SOLUTION.—By the formula,
$$\text{H. P.} = \frac{p\,W\,V}{33{,}000\,y} = \frac{2\tfrac{1}{2} \times 5.2 \times \frac{20}{60} \times 120 \times 220}{33{,}000 \times .60} = 5.78 \text{ H. P.} \quad \text{Ans.}$$

ECONOMIZERS

CONSTRUCTION AND INSTALLATION

64. According to good authority, the gases going to the chimney carry off, on an average, from 15 to 50 per cent. of the heat units contained in the fuel. By using economizers, some portion of this heat is made available for heating the feedwater; frequently, the feedwater may be heated nearly to the temperature of high-pressure steam, making a saving in some instances of 30 per cent. The heating surface of the economizer intercepts and absorbs a large percentage of the heat from the gases passing to the chimney. The

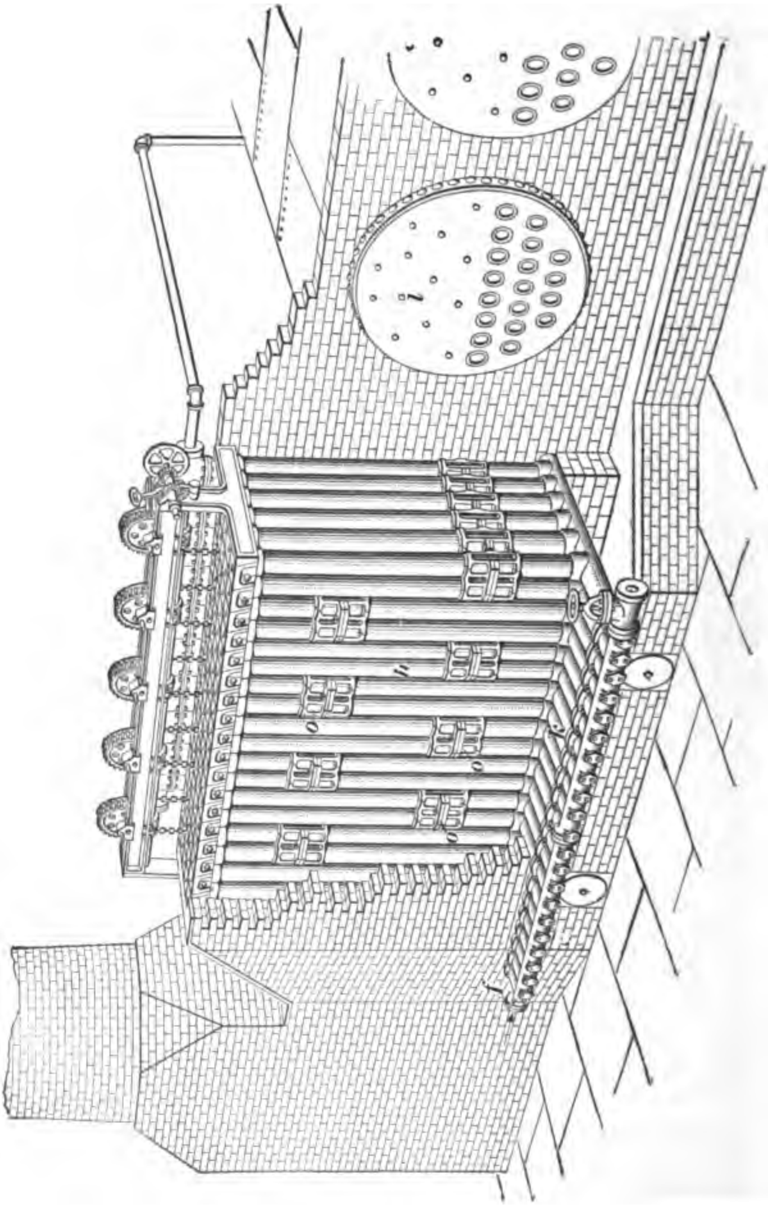


FIG. 29

more wasteful the boiler, the greater is the benefit derived from the economizer; but for large plants, it is always a valuable adjunct, and particularly where condensing engines are used. In many cases, water that has already been heated by exhaust steam may be raised to a much higher temperature by being passed through an economizer.

65. Fig. 29 illustrates a fuel economizer consisting of groups of vertical tubes *h* connected into headers *k* and so arranged that the water may be forced through the tubes in a direction opposite to the flow of the hot gases circulating around them. Thus the water enters at *f* and the coolest gases meet the coolest water; as the water becomes heated

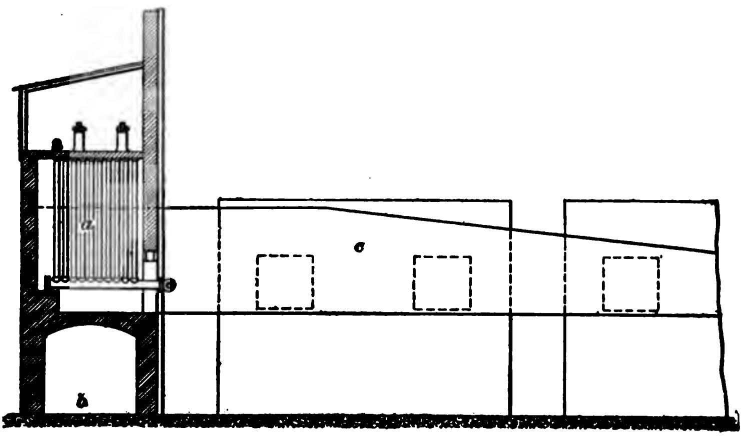


FIG. 30

by passing to the right, it comes in contact with the hotter gases nearer the boiler. The economizer is placed in an enclosing chamber, usually of brick, through which the gases pass on their way from the boilers *l* to the chimney.

Fig. 30 shows a typical arrangement of an economizer *a*, with a by-pass flue *b* for use in case it is desired to shut off from the economizer the gases that pass through the smoke flue *c* at the rear of the boilers.

66. No economizer is complete without some device for cleaning the soot and ashes from the tubes, because a coating of soot retards the absorption of heat from the gases by the water. This cleaning is effected by scrapers *o, o*, Fig. 29, that are slid slowly up and down the tubes by means of suitable gearing. Cast-iron pipes are used to avoid the pitting and corrosion to which wrought-iron pipes are subject. The temperature to which an economizer can raise the feedwater depends on the temperature of the gases leaving the boilers. Where sufficient boiler heating surface is used to reduce the gases to a low temperature before their discharge to the chimney, economizers will not be of so great advantage as where less boiler heating surface is applied.

67. The economy resulting from the use of an economizer depends on the temperature at which the feedwater is supplied, and also on the temperature of the escaping flue gases from the boiler furnaces, which should be sufficient to raise the temperature of the water from 75 to 100 degrees at least. As the economizer is expected to raise the water to a temperature above 212° F., water containing scale-forming impurities should be treated with a solvent that will prevent the deposit of scale on the inner surface of the economizer tubes. As the tubes of the economizer retard the flow of the hot gases, an intense draft is necessary. If economizers are to be used with boilers operated under natural draft, the chimney should be of extra height, but with draft induced by mechanical means, that is, by fans or steam blowers, the required intensity of draft is readily obtained and controlled.

BOILER PIPING AND AUXILIARIES

BOILER- AND ENGINE-ROOM PIPING

PIPING DETAILS

PIPES

1. The **piping** of an engine and boiler plant requires that careful attention be paid to all the details as well as to the general design, not only in order to make it suitable for the purpose, but also in order to reduce the likelihood of a breakdown. The main considerations regarding steam piping are: (1) The size of the pipes; (2) the arrangement and construction of the piping system; (3) the method of providing for expansion; (4) proper drainage.

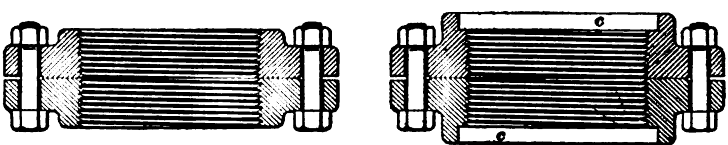
2. Most of the piping for steam and water is built up of wrought-iron pipe of standard size; but cast-iron pipe is used to some extent where the pressures are low and the increased weight is not objectionable. Ordinarily, pipe comes in 16-foot lengths, each length being threaded at both ends. Very short pieces of pipe with threads on both ends are called **nipples.** A *close nipple* is one so short that the threads cover the whole outside; that is, it is only as long as is necessary to have a complete thread at each end.

3. The various grades of wrought-iron pipe are known as *standard*, *extra strong*, and *double extra strong*. Standard wrought-iron pipe is tested at the mills by hydrostatic pressure to 300 pounds per square inch in sizes up to 1¼ inches, and to 500 pounds per square inch in sizes larger than 1¼ inches. Commercial piping in sizes of 12 inches and under with perfectly welded joints has been tested up to 1,500 pounds per square inch without bursting; 8-inch and 10-inch piping, taken at random from stock, has withstood tests of 2,000 pounds per square inch; 16-inch pipe has been tested to 800, and 24-inch pipe to 600 pounds without rupture or fracture. It would appear, therefore, that there is no reason why, for diameters less than 15 inches and pressures up to 200 pounds per square inch, piping heavier than standard should be used for the live-steam system in a power plant.

Both wrought-iron and steel pipe are used in the piping systems of power plants. Formerly, wrought iron was chiefly used, but of late steel has been employed, especially for the larger sizes of steam pipes, and has proved so reliable that, by many engineers, it is now preferred to wrought iron. The two kinds are equally reliable when made into expansion bends, copper bends being used only for very heavy work. It has been found, however, that steel water pipes corrode more easily than wrought-iron pipes, and therefore the latter is frequently preferred for water piping. All pipes, valves, and fittings used in first-class work may now readily be obtained to withstand a working pressure of 200 pounds per square inch.

PIPE FITTINGS

4. Pipe Flanges.—In all high-pressure pipe fitting for pipe diameters of 3 inches and upwards, it is most desirable to use **flange joints**; while the first cost is slightly in



(a) FIG. 1 (b)

excess of the usual threaded fitting, the facilities offered for repairs and alterations, the freedom from leaks, and the general security are sufficient to warrant the additional cost.

TABLE I

DIMENSIONS OF EXTRA HEAVY PIPE FLANGES FOR 250 POUNDS WORKING PRESSURE

Adopted by all the Leading Manufacturers

Size Inches	Diameter of Flanges Inches	Bolt Circle Inches	Number of Bolts	Size of Bolts Inches	Length of Bolts Inches
1	$4\frac{1}{2}$	$3\frac{1}{4}$	4	$\frac{1}{2}$	2
$1\frac{1}{4}$	5	$3\frac{3}{4}$	4	$\frac{1}{2}$	$2\frac{1}{4}$
$1\frac{1}{2}$	6	$4\frac{1}{2}$	4	$\frac{5}{8}$	$2\frac{1}{2}$
2	$6\frac{1}{2}$	5	4	$\frac{5}{8}$	$2\frac{1}{2}$
$2\frac{1}{2}$	$7\frac{1}{2}$	$5\frac{7}{8}$	4	$\frac{3}{4}$	3
3	$8\frac{1}{4}$	$6\frac{5}{8}$	8	$\frac{5}{8}$	3
$3\frac{1}{2}$	9	$7\frac{1}{4}$	8	$\frac{5}{8}$	$3\frac{1}{4}$
4	10	$7\frac{7}{8}$	8	$\frac{3}{4}$	$3\frac{1}{2}$
$4\frac{1}{2}$	$10\frac{1}{2}$	$8\frac{1}{2}$	8	$\frac{3}{4}$	$3\frac{1}{2}$
5	11	$9\frac{1}{4}$	8	$\frac{3}{4}$	$3\frac{3}{4}$
6	$12\frac{1}{2}$	$10\frac{5}{8}$	12	$\frac{3}{4}$	4
7	14	$11\frac{7}{8}$	12	$\frac{7}{8}$	4
8	15	13	12	$\frac{7}{8}$	$4\frac{1}{4}$
9	16	14	12	$\frac{7}{8}$	$4\frac{1}{2}$
10	$17\frac{1}{2}$	$15\frac{1}{4}$	16	$\frac{7}{8}$	$4\frac{3}{4}$
12	20	$17\frac{3}{4}$	16	$\frac{7}{8}$	5
14	$22\frac{1}{2}$	20	20	$\frac{7}{8}$	$5\frac{1}{4}$
15	$23\frac{1}{2}$	21	20	1	$5\frac{1}{2}$
16	25	$22\frac{1}{2}$	20	1	$5\frac{3}{4}$
18	27	$24\frac{1}{2}$	24	1	6
20	$29\frac{1}{2}$	$26\frac{3}{4}$	24	$1\frac{1}{8}$	$6\frac{1}{4}$
22	$31\frac{1}{2}$	$28\frac{3}{4}$	28	$1\frac{1}{8}$	$6\frac{1}{2}$
24	34	$31\frac{1}{4}$	28	$1\frac{1}{8}$	$6\frac{3}{4}$

NOTE.—Flanges, flanged fittings, valves, etc. have the bolt holes drilled in multiples of four, so that fittings may be made to face in any quarter and holes straddle center line.

Generally, this may be provided for in the arrangement of the piping; but for great lengths that are straight, or nearly so, it is necessary to use *expansion joints.* **Expansion joints** may be made in various ways. One form, shown in Fig. 4, is called the **slip joint.** The ends *a* and *b* of the

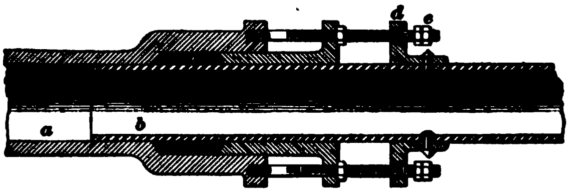


FIG. 4

sections of pipe come together in a stuffingbox *c* in order to make a steam-tight joint. The stud bolts are extra long, so as to extend through holes in a flange *d* riveted to the pipe *b*. Check-nuts *e* on the ends of the studs prevent the two ends of the pipe being forced apart by the steam pressure. The nuts *e* are not intended ordinarily to be in contact with the flange; their distance from the flange is adjusted so that the proper expansion may occur.

In Fig. 5 is shown a corrugated expansion joint, which is sometimes used on large exhaust pipes. It consists of a short section of flanged corrugated pipe, usually copper, which is put in the steam pipe wherever necessary. The elasticity of this section, due to the corrugations, permits expansion and contraction. Often the expansion is provided for by using a bent pipe usually in the form of a **U** or an **S**. Sometimes, instead of a bent pipe, one made up of fittings is used.

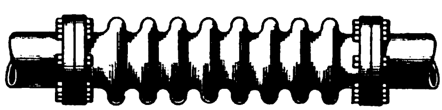


FIG. 5

7. The best way of allowing for expansion is by using bent pipes; but the space they occupy often limits their use to places where the question of space is comparatively unimportant. The forms of bends more commonly used are shown in Fig. 6, the trade name being given below each bend.

Where a bent pipe is used, the radius of the bend *r*, Fig. 6, should not be less than six times the diameter of the pipe, for wrought iron or steel; to secure the proper spring in bends used on long lines of piping, the radius should be greater than this. These bends can be used advantageously for engine connections and similar piping, as they tend to prevent vibration.

Bends of copper pipe may be of shorter radius, as copper yields more readily than iron or steel. The large sizes of copper bends are hand wrought and brazed; small sizes can be made from seamless drawn copper pipe.

8. Bends made from iron or steel pipe must be bent while red hot. Iron and steel pipe bends generally have iron flanges screwed on; copper bends either have composition flanges riveted and brazed on, as shown in Fig. 7 (*a*), or they have steel flanges, the edges of the pipe being turned over, as shown in Fig. 7 (*b*).

The piping is usually installed so that it is under a slight tension when cold; when filled with steam, the expansion of the pipes removes the tension, and there is no stress on the pipe except that due to the steam pressure.

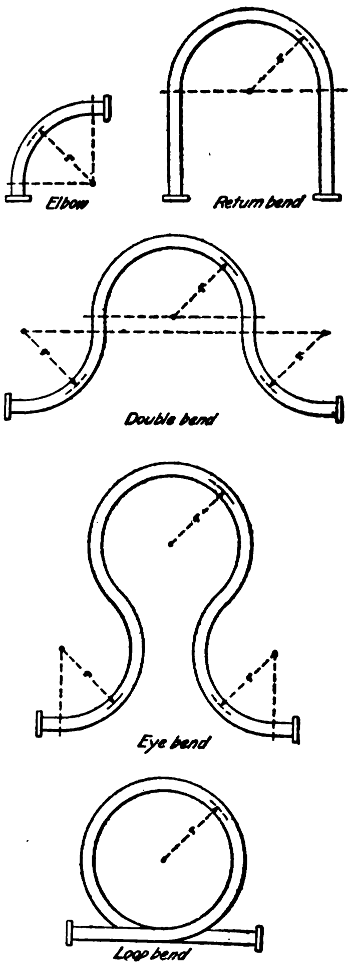


FIG. 6

9. Pipe Coverings.—To prevent loss of heat by radiation, steam pipes are covered with various kinds of materials that are poor conductors of heat. As a rule, the covering is manufactured in short sections molded in halves to fit the pipe, the valves and fittings being covered with the same material in a plastic state. After the covering is properly secured in place, it is frequently covered with a heavy duck or canvas jacket sewed on and painted, and sometimes ornamented by highly polished and lacquered brass bands placed at regular intervals along the pipes.

Fig. 7

10. The substances used for covering steam pipes for this purpose are very numerous and vary considerably in efficiency. Among the best and most widely used non-conducting materials are hair felt, cork, magnesia, asbestos, and mineral wool. Frequently, pipe coverings are made up of combinations of two or more of these substances. For pipes laid in trenches, where a cheap covering is desired, such materials as sawdust, charcoal, coal ashes, coke, loam, and slacked lime are sometimes used.

VALVES AND COCKS

11. Globe Valves.—Generally, for the purpose of controlling the flow of steam or water through pipes, one of two forms of valves is used—a *globe valve* or a *gate valve*. For steam, a globe valve is most commonly used; for water, a gate valve. Fig. 8 shows a section of a **globe valve** suitable for 150 pounds steam pressure. Inside the valve body *a*, on the partition *b*, is a flat raised seat. The spindle *c* is fitted with a disk *d* that is free to rotate on the spindle, but is caused to move longitudinally with the spindle by means of the locking sleeve *e*. The disk *d* has a recess into which is inserted a removable ring *f* of vulcanized rubber composition that can be replaced when worn out. A screw of large pitch on the valve stem permits the valve to be opened or closed quickly by turning the hand wheel. The valve should be connected to the pipe so that when closed the pressure is against the under side of the disk. This arrangement permits the stem to be packed when the valve is closed. These

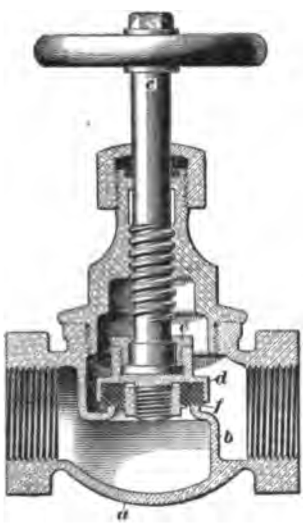


FIG. 8

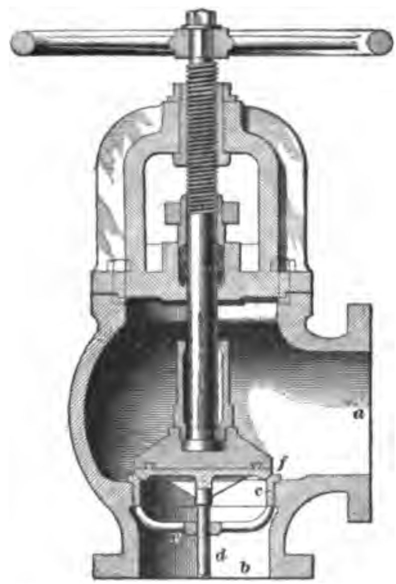


FIG. 9

valves are also made as angle valves, which sometimes conveniently take the place of elbows in the piping.

12. An **iron body valve** of the globe pattern, such as is frequently employed in piping for high-pressure work up to 250 pounds, is shown in section in Fig. 9. It is a flanged angle valve having an outlet *a* at right angles to the inlet *b*. The seat *c* is placed directly in the inlet of the valve and offers less resistance to the steam than the seat of the ordinary globe valve. Owing to the long spindle, the disk is provided with a guide *d* that moves in a hole drilled through a spider *e* cast in one piece with the brass seat. The hole in *e* is concentric with the valve seat, and the valve disk *f* is consequently guided straight to its seat.

13. Gate Valves.—Gate valves are made either as **single-gate valves,** which receive pressure on one side only, or as **double-gate valves,** which may receive pressure on either side. Some forms of double-gate valves close the opening of the valve with a solid wedge; others close with a box wedge, and others with sectional gates having either parallel or wedge-shaped seats.

Gate valves are used where but little resistance to the flow of the liquid is desired, and therefore are largely used on water and exhaust-steam connections. When they are used for steam, however, the seats should be made of bronze, which withstands high temperatures more successfully. In all gate valves, the disks rise into the upper part of the body and bonnet to allow a straight passage for the liquid.

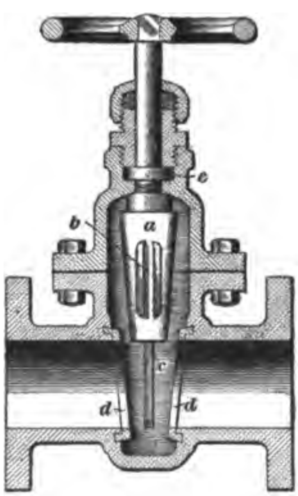


FIG. 10

14. The valve shown in Fig. 10 is a double-gate valve with a disk *a* made tapering, which is machined flat on the sides and is guided by a slot *b* in each side of the disk, fitting over a guide *c* at each side of the valve body; the disk seats against soft metallic rings *d*, *d*, firmly embedded at each side of the opening and faced off to the same taper as the disk. The valve shown is an iron-body flanged gate valve. The lower end of the stem is threaded, and the disk travels on this thread, the stem being prevented from rising by the collar *e*.

15. For valves of 6 inches and upwards on steam lines, it is desirable to use the outside-screw yoke type, with stationary wheel and rising spindle, as shown in Fig. 11. The advantages of this type are that the extension of the stem shows the position of the gate, and that the screw can always be properly lubricated and does not come in contact with the steam.

By-passes are desirable on or around all live-steam valves of 6 inches and upwards. Fig. 12 shows a gate valve provided with a small by-pass valve *a*. By first opening the

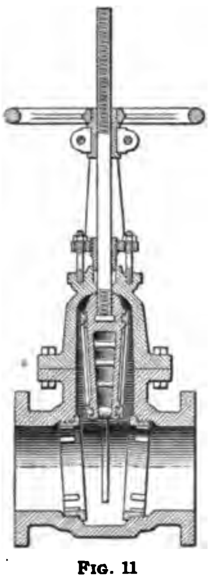
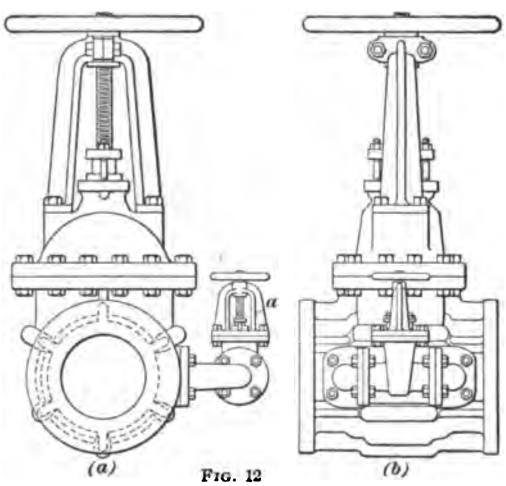


FIG. 11



(a) FIG. 12 (b)

small valve, the pressures on the two sides of the gate valve are equalized, thus making the valve easy to open.

16. Check-Valves.—**Check-valves** are valves that permit a fluid to pass through them in one direction only; they are designed so as to close automatically whenever the flow of the fluid is reversed. They are made in different forms, as vertical, horizontal, or angle check-valves.

17. Fig. 13 (*a*) shows a check-valve known as a **swing check-valve**, which may be used in either a horizontal or a vertical pipe. The valve disk *a* is attached to an arm *b* hinged at *c*. The disk and arm are so connected as to permit a slight movement of the disk so that it will seat properly. The lug *e* on the arm strikes the screw *f* when

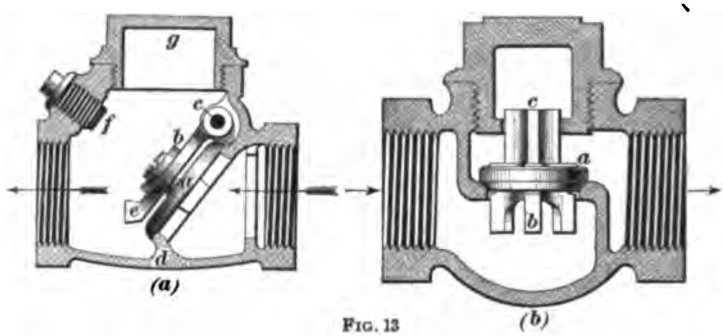


FIG. 13

swung open, thus preventing a large opening. The screw cap *g* covers the opening that gives access to the valve for inspection. The direction of flow of the fluid is indicated by the arrows. This type of check-valve probably offers less resistance to the passage of a fluid than any other form.

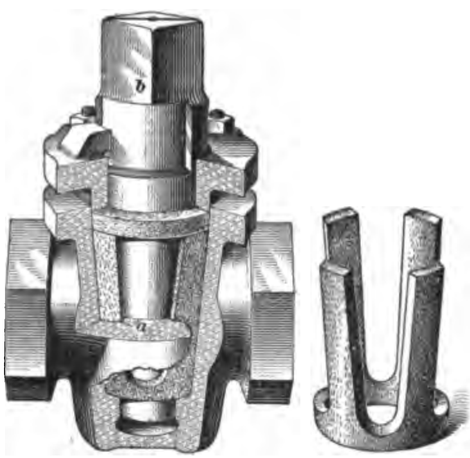

FIG. 14

18. In Fig. 13 (*b*) is shown a **globe check-valve**, the form most commonly used. The disk *a* is provided with wings *b* on the bottom and a guide *c* on the top to keep the valve from tilting sidewise, and also to prevent its opening wider than is necessary.

Special forms of these types of valve are made to take the place of elbows in pipes. In such cases, they are known as **angle check-valves.**

19. Blow-Off Valves and Cocks. Probably, no boiler fitting has given so much trouble in its operation as the blow-off valve. It is located where both internal and external corrosion is most rapid, and the action of the boiler scale flowing at a high velocity through the valve when open tends to cut out the seats. A blow-off valve should close absolutely tight to prevent leakage, and should also be capable of being easily operated. It should be of such construction as to withstand the cutting effects of scale and sediment, otherwise continual repairs will be necessary. Fig. 14 shows a good type of blow-off valve, the action of which resembles that of the globe valve shown in Fig. 9.

20. In the pipes of boiler installations, **cocks** or **plug valves** are rarely used, except in the blow-off pipe, and for this purpose they should be of special construction. Ordinary water-cocks are not suitable for this purpose, as they have a tendency to leak and it is difficult to operate them after they have been closed for some time. An improvement on the common plug cock for use as a blow-off valve is the asbestos-packed plug cock shown in Fig. 15 (*a*). Dove-tail **U**-shaped grooves *a* are cast into the interior surface of the valve body. The grooves are tightly filled with asbestos, which is elastic and makes a tight joint, and at the same time allows the plug *b* to be turned easily. For a top packing a vulcanized composition ring is used, as shown. The form of the asbestos packing contained in the dove-tail grooves in the body of the valve is shown in Fig. 15 (*b*). Since the asbestos is not affected by heat or moisture, this form of cock is very durable.



(*a*) (*b*)

FIG. 15

21. Pressure-Reducing Valves.—When steam is required at a lower pressure than that supplied by the boiler, some form of pressure-reducing valve must be used.

Such a valve is used chiefly in steam-heating systems. A reducing valve, designed to give a uniform low pressure from a varying higher pressure, is shown in Fig. 16. The steam flows through the valve from the inlet *a* to the outlet *b*, as shown by the arrows. When it is desired to use it as an angle valve, the outlet may be made at *c*. The flow of the steam is impeded and its pressure reduced by means of two disks *d* and *e* covering the ports in the interior of the valve body. These disks are connected together by the sleeve *f* and are rigidly attached to the valve stem, so that the ports are opened by the downward movement of the valve stem and closed by the upward movement. Each disk is guided by four wings on its upper side and by the valve stem, which passes through a hole in the bonnet *h*. The upper end of the stem is connected to the inner ends of two levers *i*, *i* that have their fulcrums *j*, *j* in the flange of the bonnet. The levers are pivoted on pin connections *k*, *k* in the ends of a yoke *l*. The yoke is attached to the center of a corrugated circular copper diaphragm *m*, which is subjected to the steam pressure beneath and the action of a spring *n* on top.

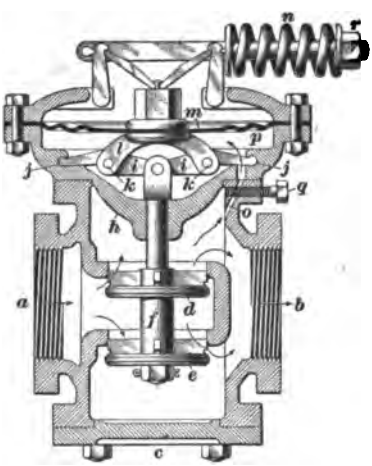


FIG. 16

The operation of the valve is as follows: When the valve is not subjected to steam pressure the ports are wide open. When steam is first admitted to the valve it passes through the ports, and some of the steam on the outlet side of the valve passes through the small steam ports *o* into the diaphragm chamber *p*, and moves the diaphragm *m* upwards until the force exerted by the steam on the diaphragm balances the resistance of the spring *n*. The amount of flow through the port *o* is regulated by means of a screw *q*. As the resistance of a spring increases very rapidly when compressed, the diaphragm, in order to counteract this increased resistance, acts on the spring through a toggle-joint, so that the diaphragm acts with an increasing leverage as it is forced upwards. The toggle-joint is so proportioned that the movement of the diaphragm is in direct proportion to the steam pressure. In order that a small movement of the diaphragm may cause a large movement of the valve disks *d* and *e*, the valve stem is not connected directly to the diaphragm, but to the longer ends of the levers *i*. When the pressure of the steam increases, it forces the diaphragm upwards, the disks *d* and *e* move toward their seats, and the area of the ports is decreased. When the pressure of the steam is reduced, the reverse action takes place.

Thus, the area of the port openings and the steam pressure on the outlet side depend on the relative position of the disks *d* and *e* in regard to their seats. This position is governed by the resistance of the spring *n*, which is regulated by a nut *r*. The valve will not only reduce the pressure, but will also regulate it automatically; that is, although the boiler pressure may vary considerably, as long as its pressure does not fall below the pressure for which the valve is set, it will give a practically uniform pressure on the discharge side. The pipe on the low-pressure side of the valve should be fitted with a steam gauge.

STEAM-PIPING ACCESSORIES

22. Separators.—A device designed to remove the entrained water, or the oil, dirt, and other impurities, from a current of steam flowing through a pipe is usually called a **separator.** When the separator is merely intended to free the steam from water, it is placed on the main pipe leading from the boiler to the engine, and as close as possible to the latter. When it is desired to remove the grease and dirt from the exhaust steam before condensing it and feeding it back to the boiler, the separator is placed in the exhaust pipe leading from the engine to the condenser.

Steam separators may be divided into two general classes—*baffle-plate separators* and *centrifugal separators.* In a baffle-plate separator, the steam comes in contact with plates generally placed at right angles to the direction of the flow of steam, the plates abruptly changing its direction of flow. In flowing through a centrifugal separator, the steam is given a whirling or rotary motion. In either case, the particles of water will continue in the original direction of the current by reason of their inertia, while the dry steam passes off in another direction.

23. The separator shown in Fig. 17 belongs to the centrifugal type. It consists of a chamber with a steam inlet and outlet, and containing a vertical pipe *a*. The steam enters by the inlet *c* and is deflected by a curved partition that gives it a spiral motion about the pipe *a*. The particles of water are thrown off by centrifugal action and run down the walls to the bottom of the chamber, while the dry steam passes through the pipe *a* and out at *d*. The separator is provided with a drain pipe *h* for the removal of the water and a gauge glass *g* for showing the amount of water that has collected. The four wings *b*, *b* are for the purpose of destroying the rotary movement of the steam after it has reached the bottom of the separator. They likewise offer additional surface to which the water particles may adhere. If it were not for these wings, the steam would keep its rotary motion while passing up the pipe *a* and thus necessarily carry some of the entrained water with it.

FIG. 17

24. The separator shown in Fig. 18 is of the baffle-plate type. The steam enters at *a* and is deflected downwards by the baffle plate *b*, which is grooved. The water dashed against the plate flows along the grooves to the sides and then to the bottom *d* of the separator. Any water not deposited on the baffle plate *b* strikes against the lower curved grating-like partition, as shown in the illustration, and falls to the bottom. The steam leaves the separator at *c*. A drain pipe *x* and stop-valve *v* are provided for draining the separator, and a glass water gauge *o* shows the amount of water that is in the bottom chamber.

FIG. 18

25. Exhaust Heads.—A special form of separator that is placed on the end of the exhaust pipe of a non-condensing engine or pump, in order to catch the water of condensation and prevent it from being scattered promiscuously over roofs, etc., in the vicinity of the exhaust outlet, is called an **exhaust head.** Exhaust heads generally serve also as **mufflers**—that is, they deaden the sound of the exhaust. They depend on changes in the direction of the current of steam for the separation of the water from the steam.

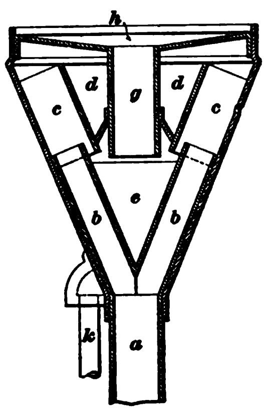


FIG. 19

Fig. 19 shows the construction of an exhaust head in common use and serves to emphasize its similarity to the separator. The exhaust steam enters through the pipe *a*, passes up through the branches *b*, *b* and nozzles *c*, *c* and strikes against the top head *h*. It then passes downwards through the cone *d*, and after making a sharp turn, escapes through *g*. The water strikes the different surfaces, drips into the collecting chamber *e*, and drains out through the pipe *k*.

BOILER-ROOM AUXILIARIES

FEEDWATER PURIFIERS AND HEATERS

PURIFICATION OF FEEDWATER

26. The necessity of adopting the most effective methods of purifying and heating water before it is fed to the boilers is now generally recognized. The preliminary purifying and heating not only saves coal and reduces the cost of cleaning the boilers, but adds materially to their life and durability.

The chemical salts held in solution and the organic or inorganic matter held in suspension in water vary in quantity, the amount depending on the source of supply of the water, the rainfall, and the season. Suspended matter forms boiler scale only by being cemented to the boiler by other materials.

It is better to remove the impurities from the water before it enters the boiler than to inject any of various compounds into the boiler to render the impurities harmless. In the latter case, the solid matter or grease must be removed from the boiler by the use of surface or bottom blow-offs and by occasionally emptying the boiler entirely and washing or scraping out the sediment. In addition to this, the accumulations of grease and solid matter inside the boiler cause a considerable loss in efficiency, and the necessity of emptying the boiler at intervals renders a larger boiler equipment unavoidable.

27. Water intended for boilers may be purified by *settlement*, by *filtration*, by *chemical means*, and by *heat*. Filtration will remove impurities in mechanical suspension, such as oil and grease, and earthy matter, but will not remove substances dissolved in the water. Chemical treatment of the water will render the scale-forming substances and corrosive acids harmless, and may be applied either before or after the water enters the boilers—preferably before, and in combination with settlement or filtration, so that no impurities may enter the boiler. Purification by heat is based on the fact that most of the scale-forming substances become insoluble and precipitate when the water containing them in solution is heated to a high temperature.

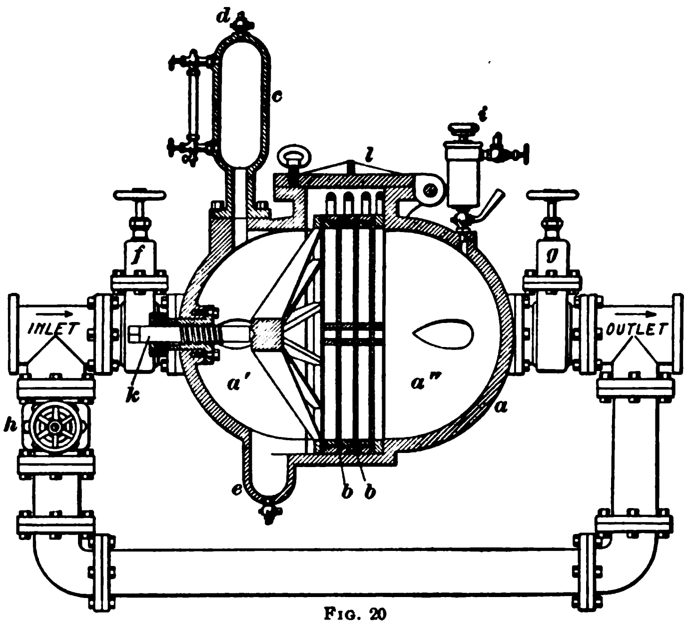


FIG. 20

28. An apparatus using cloth filters to remove oil and grease from the feedwater condensed in a surface condenser is shown in Fig. 20. It consists of a closed cast-iron vessel *a*, divided into two chambers *a'* and *a''* by perforated plates *b*, *b* covered with coarsely woven cloth. The feedwater is forced into the chamber *a'* and cannot reach the chamber *a''* except by passing through the filtering cloth. The oil and other floating impurities rise into the scum chamber *c*, whence they are removed periodically by opening the blow-off *d*.

The heavier impurities settle into the pocket *e*, which also is provided with a blow-off cock. A pressure gauge is attached to the chamber *a'*, and when this gauge indicates a pressure more than 5 pounds in excess of that in the boiler, it shows that the strainer is clogged and must be cleaned. This is done by opening the by-pass valve *h* and closing the valves *f* and *g*, thus cutting the filter out of service. The cup *i* is now filled with soda and steam is turned on, thus boiling out the filter. The soda dissolves the grease, and the impurities in the filter can then be blown out. If boiling out fails to clear the filter, the filtering diaphragms must be removed and new ones substituted. To do this, the filter is cut out of service and the feedwater is sent through the by-pass. The setscrew *k* is loosened and the hinged door *l* opened; the diaphragms can then be readily removed.

29. A chemical purification plant consists of several tanks and basins so arranged and connected by pipes that the water flows through the system by gravity. Chemicals added in the upper part of the apparatus mix with the water as it descends. The last tank is usually much larger than the others and acts as a settling tank. From this, the water is drawn off by a pump, or in some other way, to feed the boiler. The impurities and scale-forming materials remain at the bottom and are drained off after the clear water has been removed. Two settling tanks are used, so that one can supply the boilers while the impurities are settling in the other.

30. Live-Steam Purifiers.—Many of the impurities are precipitated and form scale when the water is heated to the temperature corresponding to the steam pressure maintained in the boiler. If, therefore, some device is provided in which the feedwater can be heated to, or nearly to, the temperature of water in the boiler, the impurities can be precipitated before the water is fed into the boiler. These devices might be called *combined heaters and purifiers*, but since they usually use live steam they are often called **live-steam feedwater heaters.** They are made in a variety of forms; some are constructed with removable pans for collecting the scale-forming material, while others depend on the use of a blow-off and an occasional opening of the heater for the removal of the sediment.

31. Fig. 21 shows a **Hoppes purifier,** which is of the removable-pan type. It consists of a cylindrical shell *a* fitted on one end with a removable head at *b*, the head being removed in the illustration. A series of shallow steel pans *c* are placed within the purifier. The feedwater enters at *d*

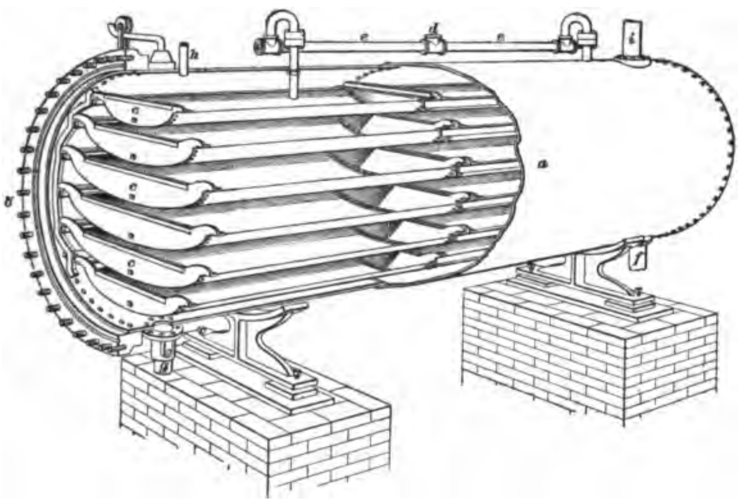


FIG. 21

and flows through the branch pipes *e*, *e* into the top pans, from which it flows in thin sheets over the edges of the pans and finally out of the purifier through the pipe *f*. Live steam from the boiler enters through the pipe *i* and heats the water in the purifier to a temperature nearly equal to its own, and in so doing precipitates those impurities that become insoluble by heat. Mud and earthy matter are deposited on the inside of the pans, while the scale-forming substances coat the outside; the pans are removed occasionally and cleaned. The feedwater flows through *f* into the boiler by gravity, the purifier being placed higher than the boiler, and having within it a pressure equal to that in the boiler. A blow-off pipe is connected at *g* and a glass water gauge connects to *g* and *h*.

32. When the boiler feedwater is highly impregnated with incrusting materials, such as carbonates and sulphates of lime and magnesia, it may be decided to use a live-steam purifier; but such a heater should be used only as a last resort, since it is more economical to use sources of waste heat, if possible, for heating the feedwater.

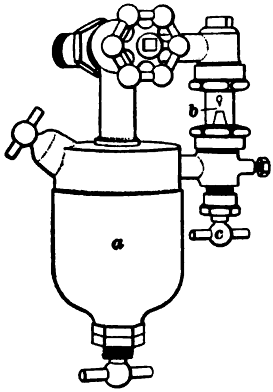


FIG. 22

When the water contains in solution such ingredients as can be neutralized or rendered harmless by the addition of such substances as caustic soda, kerosene, or trisodium-phosphate, it is desirable to add the neutralizing agent before the water enters the heater. The supply may be stored in a barrel or tank and fed regularly in the proper proportions by a suitable appliance. Kerosene fed into a boiler has the effect of loosening scale and preventing its adherence to the tubes. It is fed in by a device very similar in appearance to the lubricators used for engine cylinders.

Fig. 22 shows a so-called *boiler oil injector*. The kerosene is contained in the reservoir *a* and passes through the glass tube *b* in drops on its way to the boiler. The attendant can therefore see the amount supplied and regulate it by means of the valve *c*.

FEEDWATER HEATERS

33. Economy of Heating Feedwater.—The feedwater furnished to steam boilers must of necessity be raised from its normal temperature to that of steam before evaporation can commence, and if not otherwise accomplished, it will be done at the expense of fuel that should be utilized in making steam. This temperature, at 75 pounds gauge pressure is about 320° F., and if 60° is taken as the average temperature of feedwater, 320 − 60 = 260 British thermal units of heat is required to raise 1 pound of water from 60° to 320°. It requires 1,151.5 British thermal units to convert 1 pound of water at 60° into steam at 75 pounds gauge pressure, so that 260 British thermal units required for heating the water represents about 22.6 per cent. of the total. All heat, therefore, that can be imparted to the feedwater before it enters the boilers is just so much saved, not only in cost of fuel but in boiler capacity. It must be remembered that of the total heat generated in the furnace of the boiler, nearly 80 per cent. is lost and cannot be converted into work on account of the low efficiency of the steam engine as a heat engine. Therefore, the waste heat must be utilized, and the question of the proper selection of auxiliary appliances to obtain from this waste the largest possible benefit becomes of first importance. The unused heat units have cost just as much per unit in fuel as the useful units; hence, in the interest of economy, the largest possible percentage of the heat units that would otherwise be wasted must be returned to the boiler and thereby utilized.

The principal sources of waste heat are: the exhaust steam from engines, the exhaust steam from pumps and auxiliary appliances, and the heat carried off by the gases passing from the furnace through the flues and up the stack. Of the several methods of recovering some of these escaping heat units, the least costly are those employing exhaust-steam heaters. Methods of making use of the waste heat in the furnace gases are more expensive, but under certain conditions are preferable.

34. Types of Exhaust-Steam Feedwater Heaters. The impurities contained in the water will largely determine the type of exhaust-steam heater to be used in any given plant. These heaters are divided into two general classes—*open heaters* and *closed heaters*. An **open heater** is one in which the water space is open to the atmosphere. In a direct-contact open heater, the exhaust steam comes in contact with the water, which, by means of some one of a number of suitable devices, is broken into spray or thin sheets so that it will readily absorb the heat of the steam. In a coil heater, the exhaust steam passes through coils of pipe submerged in a vessel containing the water to be heated, and open at the top. **A closed exhaust-steam feedwater heater** is a heater in which the feedwater is not exposed to the atmosphere, but is subjected to the full boiler pressure. The steam does not come in contact with the water; the latter is heated through contact with metallic surfaces, generally those of tubes, that are heated by the exhaust steam.

When the boiler feedwater is free from acids, salts, sulphates, and carbonates, so that no scale is formed at a high temperature, the closed feedwater heater will be found satisfactory. Heaters of the coil type may be used with pure water, but should not be used with water that will precipitate sediment or scale-forming matter of any kind. The coil heater is very efficient as a heater, as the water circulating through the coils is a long time in contact with the surface surrounded and heated by the exhaust steam. Heaters of the closed type, with straight tubes and sediment chamber, can be cleaned more readily than those having curved tubes, but the curved tubes allow more freedom for expansion and contraction. Heaters of the tubular type should have ample sediment chambers and may be used with water that contains organic or earthy matter, but not with water containing scale-forming ingredients. Carbonate of lime is likely to combine with earthy matter and form an exceedingly hard scale.

Heaters of the open exhaust-steam type have the advantage of bringing the exhaust steam in direct contact with the feedwater; some of the exhaust steam is condensed, thus effecting a saving in feedwater, and sediment and scale-forming ingredients (except sulphates of lime and magnesia) are precipitated or will settle to the bottom of the heater. The oil in the exhaust steam must be intercepted by special oil extractors, filters, or skimmers, generally combined with the heater and, by automatic regulation, sufficient fresh feedwater must be added to make up the total quantity required. When the system is properly arranged, all live-steam drips and discharge from traps are led to the heater.

35. Open Exhaust Feedwater Heater.—One form of open feedwater heater is shown in section in Fig. 23. The cold water is admitted to the heater through a valve *a* and inlet pipe in the center of the upper head to a spraying device *b*, located immediately above the pans *c* that distribute the water evenly in a very fine spray. From the inner edge of the upper pan, the water falls to the pan below, and over the outer edge of this pan to the one beneath, and so on through the whole series. The pans are arranged so as to give the water a zigzag travel. Two thin cylindrical sheets of water are maintained, one at the inner edge and one at the outer edge of the pans.

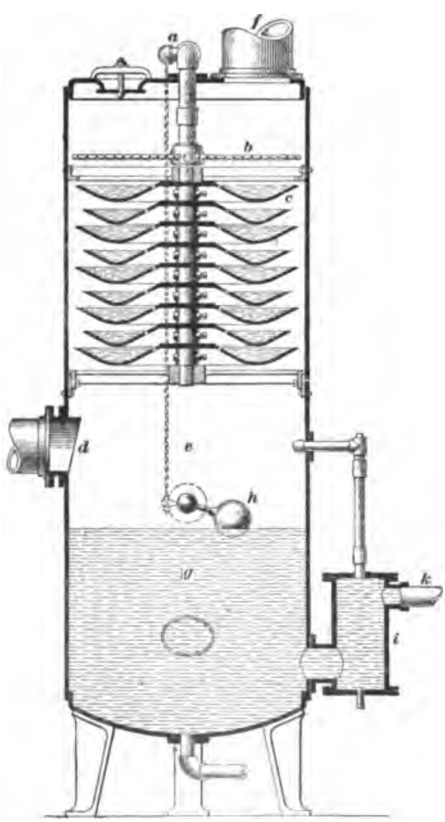


FIG. 23

The exhaust steam enters the shell through a pipe at *d* immediately below the distributing pans, and rises slowly from the large steam space *e*, coming in direct contact with the thin sheets of falling water. The steam not condensed by the cool water passes into the

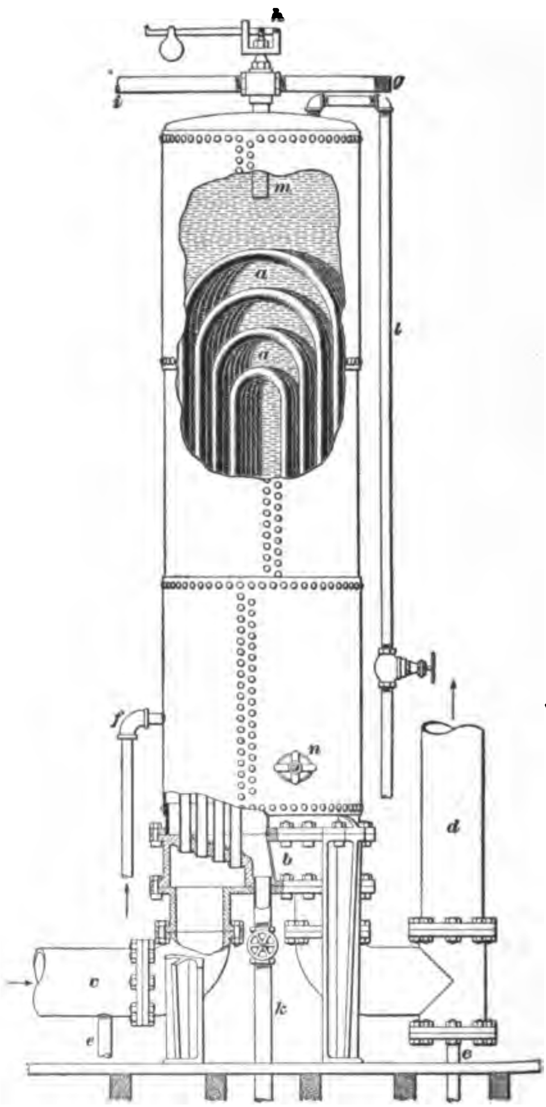


FIG. 24

atmosphere through the outlet *f* on top of the heater. As the feedwater is spread into large thin sheets, it readily absorbs the heat of the exhaust steam until the highest temperature attainable, 212° F., is reached. The purification of the feedwater also takes place to some extent in the heater, the water depositing in the pans the impurities precipitated at 212° F. The water falls from the lower pan to the large settling reservoir *g* in the lower part of the shell.

The water level is maintained automatically in *g* by means of a float *h*, which operates the valve *a* in the cold-water supply pipe on the top of the heater. As the water becomes low, the float falls and opens the valve, thus admitting more water until the rising of the float closes the valve.

The oil and lighter impurities float on top of the water and are thus prevented from entering the chamber *i*, from which the water is taken into the boiler. The chamber *i* is connected to the water reservoir by a large flanged connection at the bottom and by a smaller pipe to the steam space of the heater. The feed-pump takes its water supply through the suction pipe *k* attached to the upper part of the small chamber *i*. The object of connecting the top of this chamber with the steam space *e* is to prevent the water level in the reservoir *g* from falling below the level of the pump connection *k*. As soon as it falls to the level of the pump connection, steam from the heater will flow into the pump, thus preventing the pump from taking any more water from the heater.

36. Closed Exhaust Feedwater Heater.—In Fig. 24 is shown one form of closed exhaust-steam feedwater heater, in which the cool water is outside of the tubes and the steam inside. The heating surface consists of inverted U tubes *a*, through which the exhaust steam passes. The ends of the tubes are expanded into holes in the cover-plate of the bottom compartment *b* of the heater, which has a vertical partition across the middle. As both ends of the tubes are expanded into the same tube-sheet, in the way shown in Fig. 25, the joints are not affected by the expansion or contraction of the tubes. The exhaust steam enters one side of the compartment *b* through the pipe *c* and flows upwards through one leg of the tubes, down the other, and out of the pipe *d* to the atmosphere.

The condensed steam is discharged through small pipes *e* at the bottom of the two parts *c* and *d* of the exhaust pipe. The feedwater enters the shell of the heater through a pipe *f* near the bottom, surrounds the tubes *a*, and passes out through the pipe *g* attached to the center of the top head of the heater. A safety valve *h* is fitted to the heater to prevent excessive pressure due to closing the valve in the feedpipe before the pump is stopped. When the safety valve is open, the water discharges through the overflow pipe *i*. The heater is provided with a bottom blow-off pipe *k* and a surface blow-off pipe *l* for removing from the water both settled and floating impurities. To prevent the floating impurities from entering the feedpipe *g*, it extends several inches below the surface of the water, as shown at *m*. A handhole *n* is provided for removing scale. A heater of this form will raise the temperature of the water to 200° or 212° F. and precipitate most of the scale-forming substances therein.

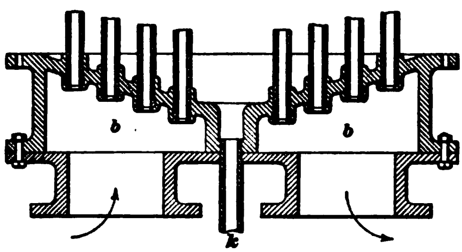


FIG. 25

STEAM TRAPS

PURPOSE AND CLASSIFICATION

37. The water of condensation in steam piping must be allowed to escape freely, both in order to admit uncondensed steam and also to prevent an accumulation of water that may cause *water hammer*. The use of an open pipe for this purpose is very objectionable, as it will allow steam to escape as well as water; to prevent the escape of steam and at the same time allow the water to escape freely, devices known as **steam traps** are employed.

Steam traps are divided into two general classes, known respectively as *open* or *discharge traps*, and *return* or *closed traps*, the discharge pipe of the former being open to the atmosphere, while the latter returns the water of condensation directly to the boiler.

38. Open or Discharge Trap.—Fig. 26 shows one form of the open trap. The bucket *a* floats in the water, holding the end of the spindle *b*, which is made with four

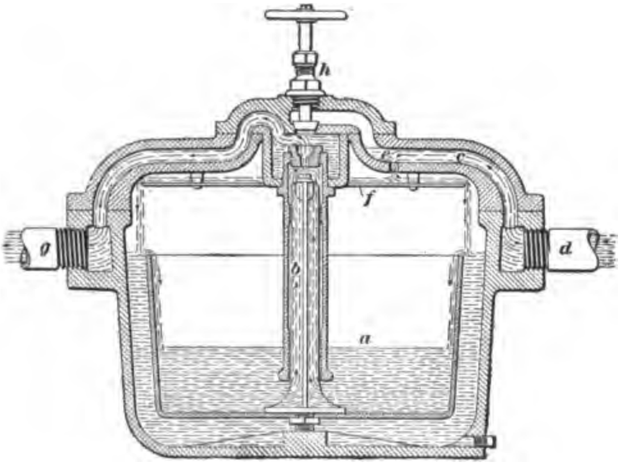


FIG. 26

wings, against the seat in the cover. The water entering the inlet chamber *c* from the inlet *d* passes through an opening *e* into the body of the trap. A diaphragm *f* is placed beneath the opening to diffuse the water toward the sides, so that the water will fill the body and overflow into the bucket, which, by falling, opens the discharge valve. The water in the bucket is forced by the steam pressure through the sleeve, around the spindle, and through the discharge chamber, into the discharge pipe connected at *g*. A by-pass *h* is provided at the top to allow steam to be blown through the discharge pipe.

39. Steam-Trap Connections.—A good method of connecting up a steam trap is illustrated in Fig. 27. The drip pipe *a* and the discharge pipe *d* are connected to the trap with unions for the purpose of easily disconnecting the trap *e*. A by-pass is formed by the pipe *b* connected to the discharge pipe *d*. When it is necessary to allow the steam to pass by the trap, the valves in the pipes *a* and *c* are closed and that in the pipe *b* is opened. To allow the trap to discharge the water freely, the waste pipe *d* should be at a lower level, unless the pressure within the trap is sufficient to lift the water discharged by it to a level higher than that of the trap; but the pressure on the discharge pipe must be less than that in the trap, as otherwise it will not operate.

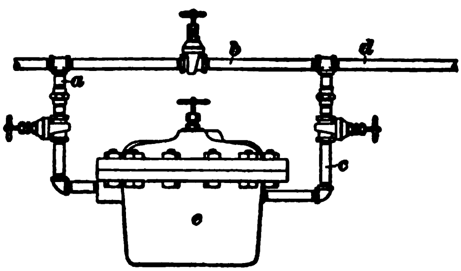


FIG. 27

40. Closed or Return Steam Trap.—A return trap must be located at such a height above the water in the boiler that the hydrostatic head produced by the elevation of the trap will cause the water of condensation to flow therefrom into the boiler. One form of return trap, with the necessary piping and valves to connect it to a boiler, is shown in Fig. 28. It consists of a cast-iron receiver *a* supported at one end by hollow trunnions *b* and *c* on the stationary part of the trap and at the other end by a link, lever, and weight *g*, as shown. In the drainage of a heating or steam-pipe system, the different return pipes lead to a tank, as shown at *d*. The water rises through the pipe *e*, passing through the check-valve *f* and trunnion *c* to the receiver *a*. The water enters this receiver until its weight is sufficient to overbalance the counterweight *g*, when the receiver *a* moves downwards until it comes against the guide *h*. This downward motion causes the lug *i* to engage the upper nuts on the stem of the steam valve *j*, opening the latter and thus admitting steam at full boiler pressure on top of the water. The steam enters from the boiler through the pipe *k*, trunnion *b*, and curved pipe *l*, leading to the highest point in the receiver. Driven by the steam, the water flows

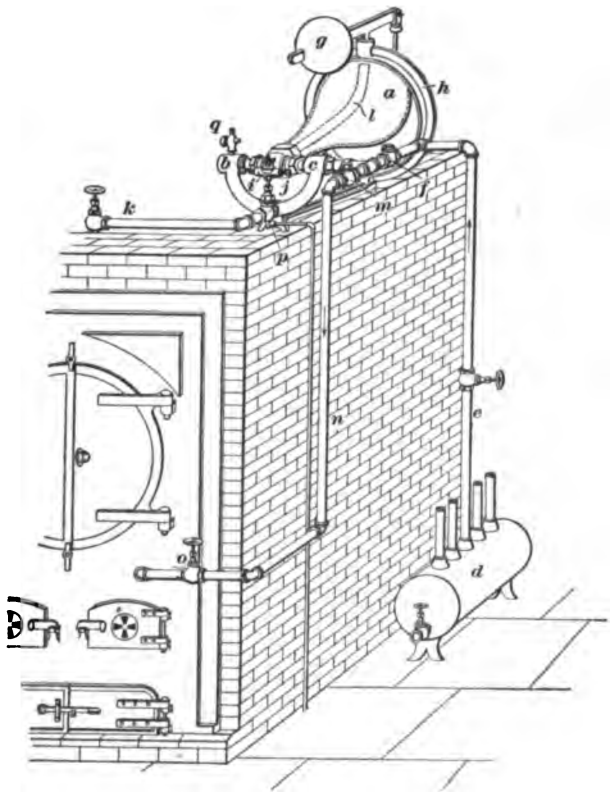


FIG. 28

from the receiver to the boiler by gravity, through the trunnion *c*, check-valve *m*, pipe *n*, and globe valve *o*. As soon as the receiver is emptied, the weight *g* lifts it to its upper position, which closes the steam valve *j* and opens a small air valve *p* below the valve *j*, allowing the steam to exhaust from the receiver. A cock *q* is provided on the trunnion *b* for the purpose of venting the interior of the receiver by hand, if necessary at any time.

41. When there is not sufficient pressure to make the water in the receiver enter the trap on top of the boiler, another trap may be placed at the point where water will flow into it. This trap may then be made to discharge into one placed on top of the boiler, using steam from the boiler as a motive force.

Return traps can be made to discharge the water into elevated tanks, the height to which the water may be raised depending on the available boiler pressure. This height, in feet, allowing for frictional and other resistances, is given approximately by multiplying the boiler pressure available by 1.4. Thus, if the boiler pressure is 60 pounds per square inch, a return trap can discharge into a tank $60 \times 1.4 = 84$ feet above it.

DESIGN AND ARRANGEMENT OF PIPING

PRINCIPLES OF DESIGN

GENERAL REQUIREMENTS

42. The installation of a complete steam plant includes the setting of the boiler or boilers, the arrangement of the various lines of piping, and the location and arrangement of the various accessories, such as feedwater heaters, purifiers, separators, economizers, feed-pumps, and injectors. An elaborate plant may be fitted with economizers, mechanical stokers, coal conveyers and ash conveyers, purifiers, and other labor-saving and fuel-saving devices. On the other hand, the plant may consist simply of boilers, chimney, and feed-pump.

The proper designing of a system of piping requires a careful analysis of the conditions of service, a thorough knowledge of the methods of distributing and conveying steam and water, and of the quality and strength of materials employed. A system of steam piping for a power plant must be so designed as to insure reliability of service and economy of construction. The main lines of piping should be so connected that it will not be necessary to shut down the entire plant to make minor repairs. This continuity of operation is absolutely indispensable to a successful power plant.

The pipes and fittings must be so proportioned as to permit a free flow of steam or water, so that no undue loss will be caused by condensation, radiation, or friction. The steam piping should be so arranged that water pockets will be avoided; and where such pockets are unavoidable, they must be drained to free them from water; the entrained water can be automatically returned to the boiler. By-pass pipes, with suitably placed valves, should be arranged around feedwater heaters, economizers, pumps, etc. The system must be so designed as to give perfect freedom for expansion and contraction, without undue stress on any part of the system, and without opening joints and thus causing leakage.

43. Perfect drainage must be provided in order that all water of condensation shall be fully separated from the steam, and, by suitable traps or return systems, delivered again to the boiler. An elaborate duplication of steam mains and connections is not necessary. The double, or duplicate, system of piping was introduced to overcome the deficiencies in valves, fittings, and methods of workmanship, and to insure greater reliability; but this method has no further reason for application, because manufacturers can now furnish everything necessary for a first-class system. Reliability is insured by careful design and superior workmanship, combined with the use of high-class materials and fittings and the judicious placing of cut-out and by-pass valves.

Drainage is best effected by arranging the piping so that all the water of condensation will flow by gravity toward a point close to the delivery end of the pipe, and then providing a drip pipe at that point. In the case of large pipes, a trap may be placed at the end of the drip pipe for automatic draining; the trap serves to seal the end of the drip pipe and thus prevents waste of steam.

The presence of water in a steam pipe is the cause of **water hammer**, the term used to describe the condition that causes the hammering noise often heard in the piping of steam-heating plants. It has been shown experimentally that the pressure produced by water hammer may be as great as ten times that which the pipe is expected to sustain in its regular work. In some cases, water hammer has caused boiler explosions by bursting a steam pipe and thus relieving the boiler pressure so suddenly that a large quantity of water flashed into steam and burst the boiler.

CONDITIONS AFFECTING DESIGN

44. When steam leaves the boilers and starts to flow through a pipe of a given diameter, several factors tend to change the form of the original energy possessed by the steam; among them are *condensation* and *friction*.

Condensation is of two kinds—*static condensation*, which occurs when the steam fills the pipe, but is not flowing through it, and *dynamic condensation*, which takes place when a valve is opened, permitting the steam to flow through the pipes. The latter should be less than the former, because of the fall in pressure and temperature that takes place at the delivery end of the pipe, and the effort that is being constantly made to raise or maintain the original condition of the steam; but it is often found that the amount of condensation is very nearly equal in both cases.

Friction in the pipe causes a loss of pressure and requires work to be done to overcome the loss. The natural condition of condensation, combined with the loss in pressure due to friction, cannot be wholly overcome; no matter to what extent the piping is covered with the best kinds of non-conducting covering, there is still a loss of heat.

45. Friction is greater through elbows of short radius than through elbows of long radius. Globe valves offer a serious impediment to the passage of high-pressure steam, but the drop in pressure when passing through a gate valve is practically negligible. The loss of head due to the friction of the steam entering the pipe and passing elbows and valves will reduce the flow below the estimated capacity of straight pipes. The resistance at a globe valve is usually assumed to be about the same as that for a length of pipe equal to sixty diameters, while the resistance at an elbow is assumed to be approximately equal to two-thirds that of a globe valve. It is assumed that the resistance at entrance is equal to the resistance offered by a globe valve, or sixty diameters. These equivalent lengths of straight pipe must be added to the actual length of the pipe, if the loss due to friction is to be calculated closely.

The rigid requirements of reliable and continuous operation of power plants have, within the past few years, revolutionized the simple methods of piping of earlier days, and have led to great improvements in the design and quality of all the materials and fittings essential to a complete system.

FLOW OF STEAM IN PIPES

46. Flow of Steam Under Pressure.—The amount of steam that will flow continuously through a pipe of given diameter in 1 minute at specified pressures may be calculated by the following formula:

$$W = 87\sqrt{\frac{w\,(p_1 - p_2)\,d^5}{L\left(1 + \frac{3.6}{d}\right)}} \qquad (1)$$

where W = weight of steam discharged, in pounds, per minute;

w = weight of steam per cubic foot, at initial pressure p_1;

p_1 = initial pressure;

p_2 = final pressure;

L = length of pipe, in feet;

d = diameter of pipe, in inches.

The difference $p_1 - p_2$ is equal to the drop in pressure in the pipe. If the drop in pressure through the pipe is considered to be 1 pound and the length L to be 100 feet, then

$$W' = 87\sqrt{\frac{w\,d^6}{100\left(1+\frac{3.6}{d}\right)}} = 8.7\sqrt{\frac{w\,d^6}{d+3.6}} \qquad (2)$$

where W' = weight, in pounds, of steam delivered per minute through a 100-foot length of pipe.

EXAMPLE.—How many pounds of steam per minute can be delivered through 100 feet of 3-inch pipe with an initial pressure, as shown by the steam gauge, of 100 pounds, if the loss of pressure in the pipe is 1 pound?

SOLUTION.—If the gauge pressure is 100 lb. per sq. in., the absolute pressure must be $100 + 14.7 = 114.7$. A cubic foot of steam at this pressure will weigh, approximately, .258 lb.; d = 3 in., and $d^6 = 729$. Then, substituting in formula **2**,

$$W' = 8.7\sqrt{\frac{.258 \times 729}{3+3.6}} = 46.4 \text{ lb. per min. Ans.}$$

47. In applying the foregoing formula in determining the drop of pressure in, or the size of, the supply pipe of steam engines in an attempt to base the size of this pipe on the rated horsepower and steam consumption per horsepower per hour, the fact that, in most cases, the steam is taken intermittently from the pipe must be considered. Thus, assume that an engine of 100 horsepower, consuming 35 pounds of steam per horsepower per hour, cuts off at one-fourth stroke. In that case, the steam consumption per hour is $100 \times 35 = 3{,}500$ pounds. But as steam is taken for each stroke only during one-quarter of the time consumed for the stroke, the 3,500 pounds of steam flow through the pipe in $\frac{1}{4}$ hour. Then, in order to obtain the quantity of steam that would flow continuously at the same velocity at which it flows up to the point of cut-off, the actual steam consumption per hour should be divided by the fraction representing the point of cut-off and the quotient taken as the weight of steam to be used in calculating the drop of pressure or the size of pipe. Thus, in the case mentioned, the weight of steam to be used in the formula will be $3{,}500 \div \frac{1}{4} = 14{,}000$ pounds per hour.

When the flow of steam in a steam pipe is constant, as, for instance, in the supply pipe used for a direct-acting steam pump, the weight of steam to be used for purposes of calculation is equal to the actual weight of steam used per hour.

The foregoing formulas for the flow of steam through pipes, like all similar formulas, give approximate results only, but the results are near enough for practical purposes. The actual inside diameter of pipe is very rarely the same as the nominal diameter, and it is seen from formula **1** that a slight change in the diameter has a great influence on the flow of steam.

TABLE III

WEIGHT OF STEAM DELIVERED PER MINUTE THROUGH 100 FEET OF PIPE FOR STANDARD PIPE SIZES AND VARIOUS INITIAL PRESSURES

Initial Pressure by Gauge	Nominal Inside Diameter of Pipe, in Inches										Initial Pressure by Gauge
	3	3½	4	4½	5	6	7	8	9	10	
	Weight, in Pounds, of Steam Delivered per Minute Through 100 Feet of Pipe With 1 Pound Loss of Pressure										
70	43.2	64.5	91.7	124.3	168.7	277.2	410.5	577.2	793.3	1,051.7	70
80	45.5	68.0	96.6	130.9	177.7	292.1	432.5	608.2	835.8	1,108.4	80
90	47.6	71.2	101.2	137.2	186.3	306.0	453.3	637.3	875.9	1,161.3	90
100	49.7	74.3	105.7	143.2	194.4	319.8	473.0	665.1	914.1	1,211.8	100
110	51.7	77.3	109.9	148.9	202.1	332.2	491.8	691.5	950.4	1,259.9	110
120	53.6	80.2	113.9	154.4	209.5	344.3	509.9	717.0	985.4	1,306.3	120
130	55.4	82.9	117.8	159.7	216.7	356.1	527.4	741.6	1,019.6	1,351.1	130
140	57.2	85.5	121.5	164.7	223.6	367.4	544.6	765.7	1,052.9	1,393.9	140
150	58.9	88.1	125.2	169.6	230.2	378.3	560.2	787.7	1,082.6	1,428.1	150

48. Table III gives the approximate weights of steam delivered per minute through 100 feet of pipe of various diameters with a drop in pressure of 1 pound. On the whole, these values are slightly higher than those given by formula **1.** If the allowable drop in pressure is to be more or less than 1 pound, multiply the values given in the table by the square root of the drop. If the length is more or less than 100 feet, divide 100 by the length, in feet, and multiply the figures in the table by the square root of the quotient.

EXAMPLE.—How many pounds of steam will be discharged per minute, with 120 pounds initial gauge pressure, through a pipe 3 inches in diameter and 400 feet long, the allowable loss of pressure being 2 pounds?

SOLUTION.—From Table III, the amount discharged through 100 ft. of 3-in. pipe for a loss of 1 lb. is 53.6 lb. per min. for an initial pressure of 120 lb. With a loss of 2 lb. pressure the amount discharged will be $53.6 \times \sqrt{2}$. For a length of 400 ft., the discharge will be

$$53.6 \times \sqrt{2} \times \sqrt{\frac{100}{400}} = 37.9 \text{ lb. per min. Ans.}$$

Usually, it is not necessary to calculate the thickness of wrought-iron or steel pipe, because there are standard sizes that the makers have designed to be used with certain working steam pressures. Other stresses on the pipe are much greater than the ordinary working pressure, so that if the pipe is strong enough to withstand threading and the making up of joints it will be strong enough for the pressure.

49. Velocity of Flow of Steam.—In practice, the velocity of the flow of steam in the steam pipes of engines and pumps is usually not greater than 6,000 feet per minute, although in some cases this value is exceeded; occasionally it is increased to as much as 8,000 feet per minute. The calculation of velocity assumes that the cylinder is filled with steam at boiler pressure at each stroke. For exhaust piping, a common value is 4,000 feet per minute, it being assumed that a volume of steam, at release pressure, equal to the volume of the cylinder is exhausted at each stroke. In some cases, the exhaust pipe is made from 75 to 100 per cent. larger than the steam pipe. The area of the pipe may be calculated by the following formula:

Let A = area of cylinder, in square inches;
a = area of pipe, in square inches;
S = piston speed, in feet per minute;
s = velocity in pipe, in feet per minute.

Then,
$$a s = A S$$
and
$$a = \frac{A S}{s}$$

EXAMPLE 1.—What should be the area of the steam pipe of an engine having a cylinder diameter of 20 inches and a piston speed of

450 feet per minute, the velocity of the steam in the pipe being taken at 6,000 feet per minute?

SOLUTION.—Area of cylinder,

$$A = .7854 \times 20^2 = 314.16 \text{ sq. in.}$$

Then, by the above formula,

$$a = \frac{AS}{s} = \frac{314.16 \times 450}{6,000} = 23.56 \text{ sq. in., say 24 sq. in.} \quad \text{Ans.}$$

EXAMPLE 2.—What should be the area of the exhaust pipe of the engine in example 1, the velocity s of the steam being taken as 4,000 feet per minute?

SOLUTION.—

$$a = \frac{AS}{s} = \frac{314.16 \times 450}{4,000} = 35.34 \text{ sq. in., say 35 sq. in.} \quad \text{Ans.}$$

ARRANGEMENT OF PIPING

50. In arranging a piping system for a steam plant, the aim must be to produce a design that combines low first cost with durability and serviceableness. A point that must be considered is the extent to which the piping must be in duplicate in order to prevent a shut-down in case of an accident to any section. The ease with which the piping can be taken down for repairs must also be considered. In general, flanged sections are more easily taken down than sections united by screwed joints, at least in the larger sizes. When screwed joints are used, it is advisable to introduce a liberal supply of unions, so that a section may be taken out and replaced without having to tear down the whole piping system. The question whether to place the piping overhead or under the flooring is chiefly one of convenience and appearance. With the piping under the flooring, the engine room will generally look better, but the piping will not be as accessible as when overhead.

51. Careful thought is necessary in designing piping connections to boilers. Connections between a single boiler and the distributing main are comparatively simple to make, but when two or more boilers are to be connected to the same main line of pipe, special and adequate provision must be made for expansion, otherwise the stresses on the connections will cause them to leak. If a long line of pipe is connected directly to the boiler shell, the expansion due to the entrance of hot steam will so increase the length of the pipe as to twist or wrench the joints and cause them to leak.

Several approved methods of connecting up the main steam supply pipe with the boilers are shown in Figs. 29 and 30. In Fig. 29, the connections are made by means of bent pipe, while in Fig. 30 straight pipe is used. In Fig. 29 (*a*) and (*c*), a short length of pipe rises vertically above the shell of the boiler and connects with a bent branch pipe

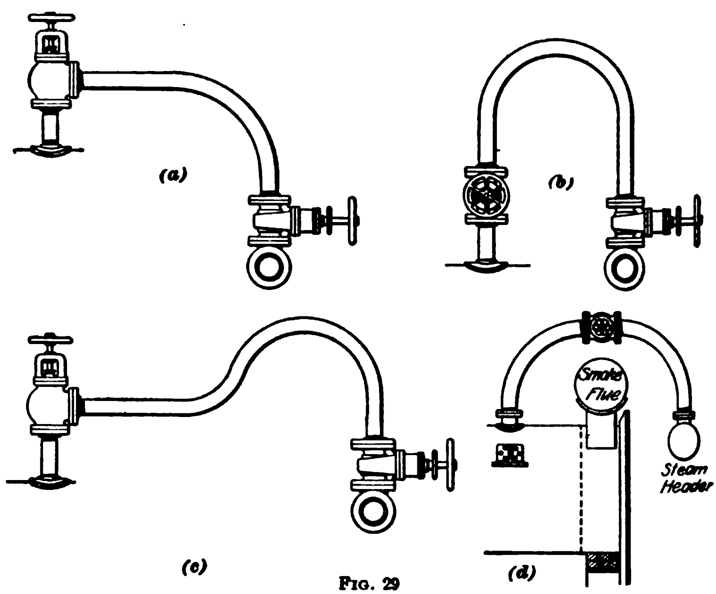


Fig. 29

joined to the main steam pipe or header, the bent pipe allowing the header to expand and contract freely. In Fig. 29 (*b*), connection between the boiler and the header is made by using a **U** bend. Fig. 29 (*d*) illustrates the use of two quarter-turn bends in making the connection. It is generally conceded that when pipe bends are thus used, the best position for the valve, when only one is used, is at the center of the bend, but some engineers regard it as better practice to use two valves, one being placed near the main and the other at the boiler. When two valves are used, it is frequently necessary to tap the body of each valve for a drip connection to drain away any water of condensation that may accumulate in it.

52. In Fig. 30, in which straight pipe is used, the length of the vertical sections should be great enough to give the spring necessary to allow expansion without straining any of the parts. In Fig. 30, *a* is the branch pipe, *b* the steam main,

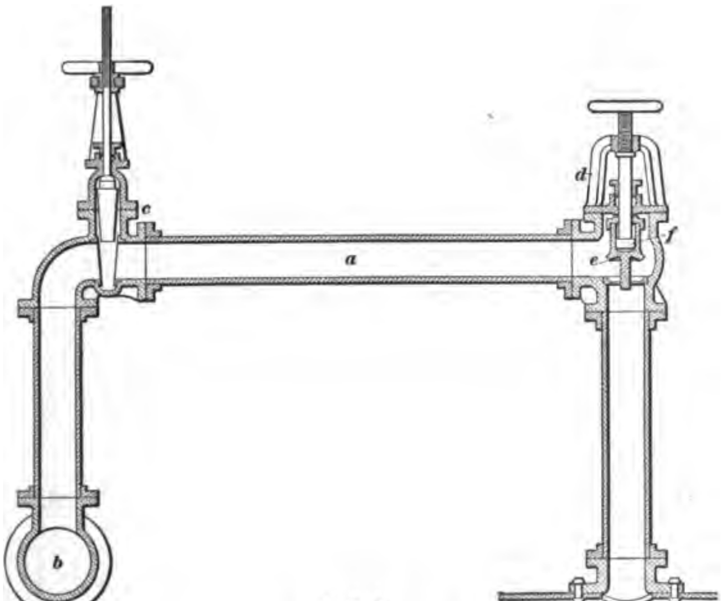


FIG. 30

or header, and *c* an angle or gate valve. In high-pressure steam plants, it is customary to insert, in addition, a non-return automatic stop-valve *d* in the branch from each boiler to the steam main. Its object is to prevent, automatically, the flow of steam from one boiler to any boiler that may be disabled. The non-return valve illustrated is similar to an ordinary angle globe valve, except that the valve disk *e* has sufficient vertical play on the lower end of the stem *f* to allow it to seat, should the pressure in the individual boiler

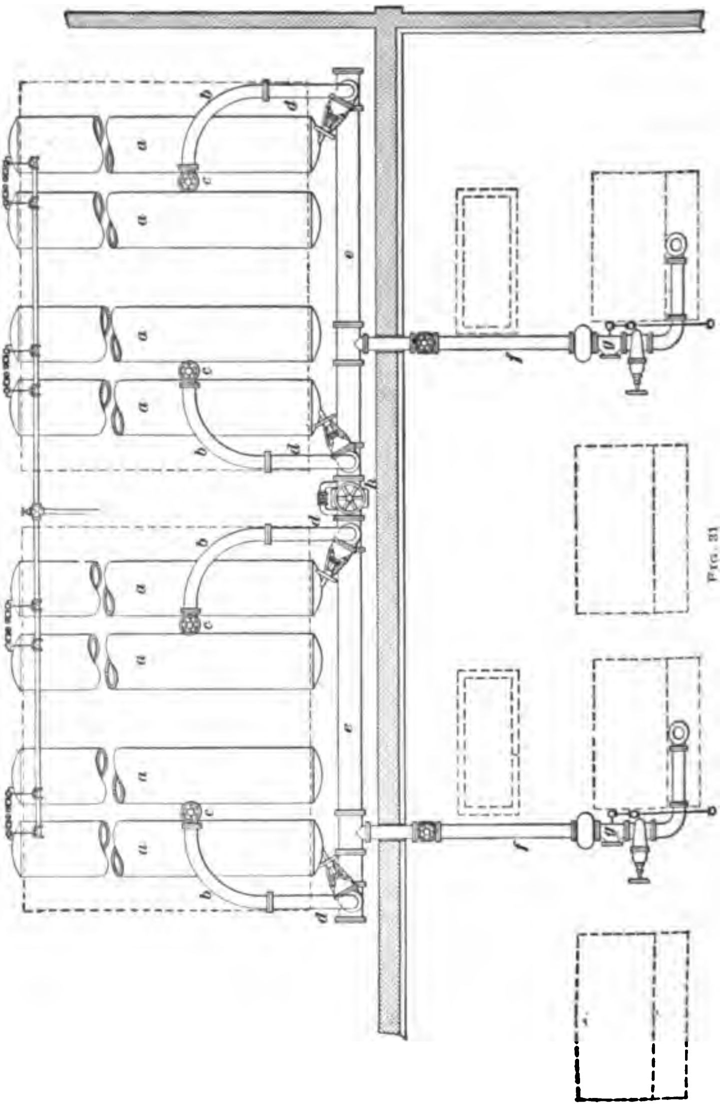


FIG. 21

become much less than that in the main, even when the valve stem is in its highest position. The arrangement is such that the disk may be firmly held to its seat when desired. Gate valves are not suitable for this kind of emergency work, as they require considerable time to close, and may be difficult to move when nearly closed.

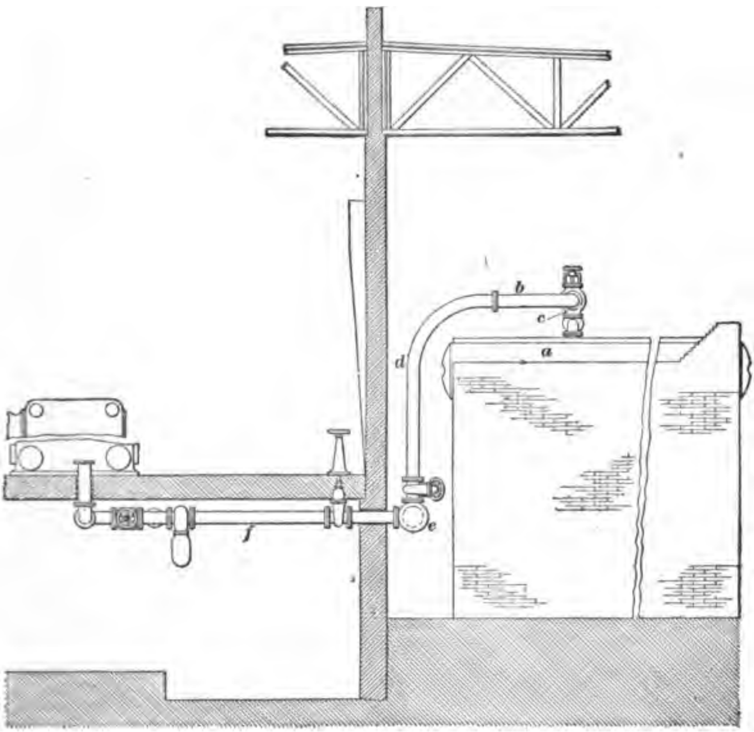


FIG. 32

53. Where several boilers of, say, 200 horsepower and upwards are used, it will be found very convenient to place the steam main or header on or near the floor in the rear of the boilers; this brings all the large valves in accessible positions. The steam lines leading to the engines are placed below the engine-room floor. This system is particularly applicable where horizontal engines are used.

The judicious use of long-radius bends, a convenient arrangement of valves, accessible location of the live-steam header and steam connections to engines below the engine-room floor are shown in Figs. 31, 32, and 33. From the cross-connection between the steam drums *a*, *a* of the water-tube boilers leads a connection *b*, starting from an automatic stop- and check-valve *c*; the long-radius bend *b* is placed horizontally and connects with a similar bend *d* leading vertically in a downward direction to the live-steam header *e*. This arrangement gives great elasticity to a system of large piping and the valves are in convenient positions for ready manipulation.

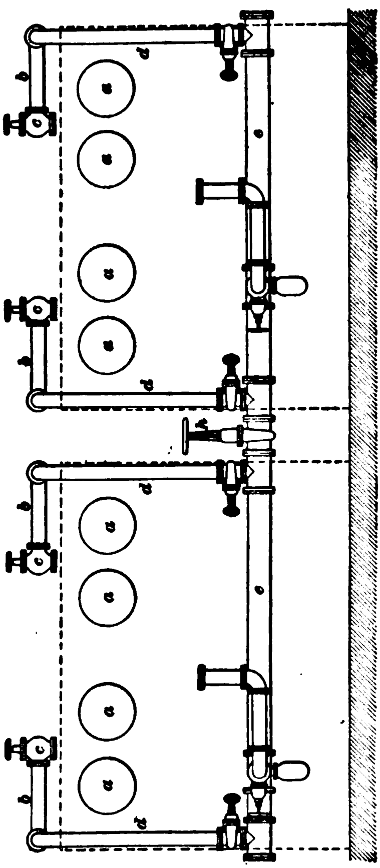
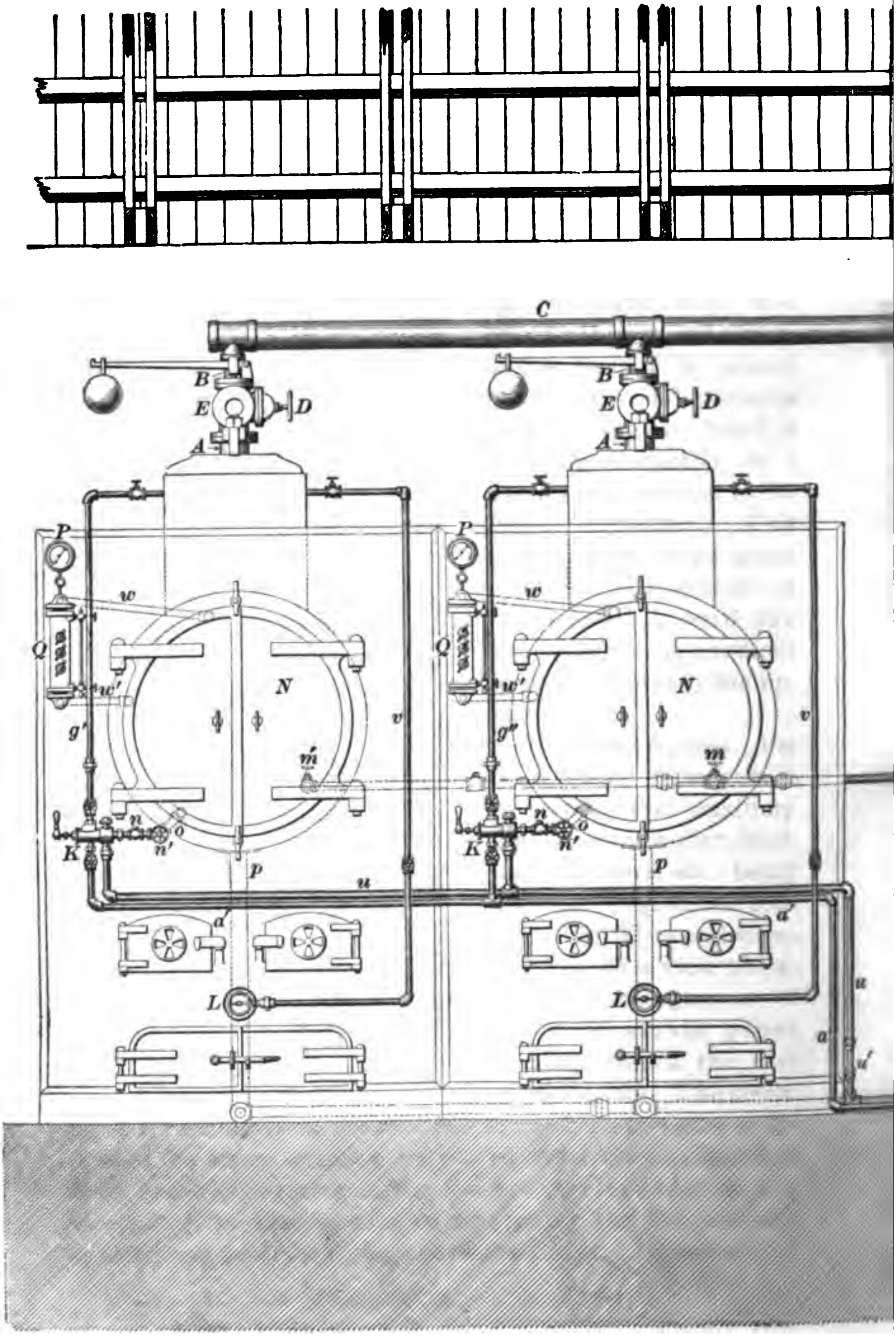


FIG. 33

In these figures, the main steam piping only is shown, the auxiliary piping for the boiler, feed-water heaters, etc. being omitted. Fig. 31 is a plan view, Fig. 32 an end view, and Fig. 33 a view showing the arrangement of the main steam pipes looking toward the rear of the boilers. The steam pipes *f*, running from the header *e* to the high-pressure cylinders of the steam engines, are placed under the engine-room floor, and a connection to the low-pressure cylinder is provided at *g*, so that in case of emergency the low-pressure cylinder can be run with high-pressure steam. By examining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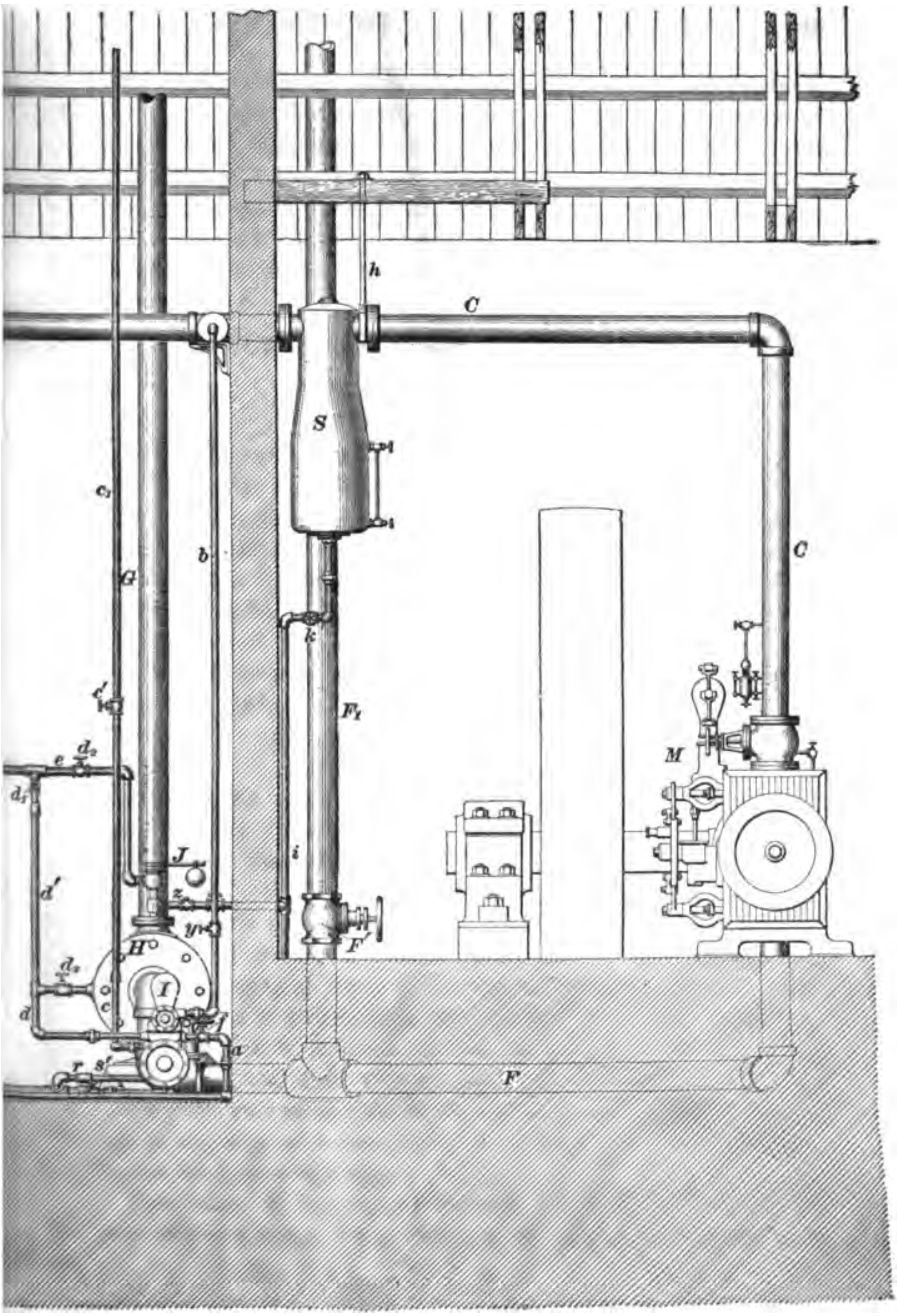
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the arrangement of valves between the boilers and engines, it will be seen that without duplicate piping it is possible to cut out any engine or boiler in case of accident and still run the plant with the remaining engines and boilers. The main steam header is divided into two sections by the large gate valve *h*, Figs. 31 and 33, so that one half of the header can be cut off from the other half.

GENERAL ARRANGEMENT OF PLANTS

MEDIUM-SIZE PLANTS

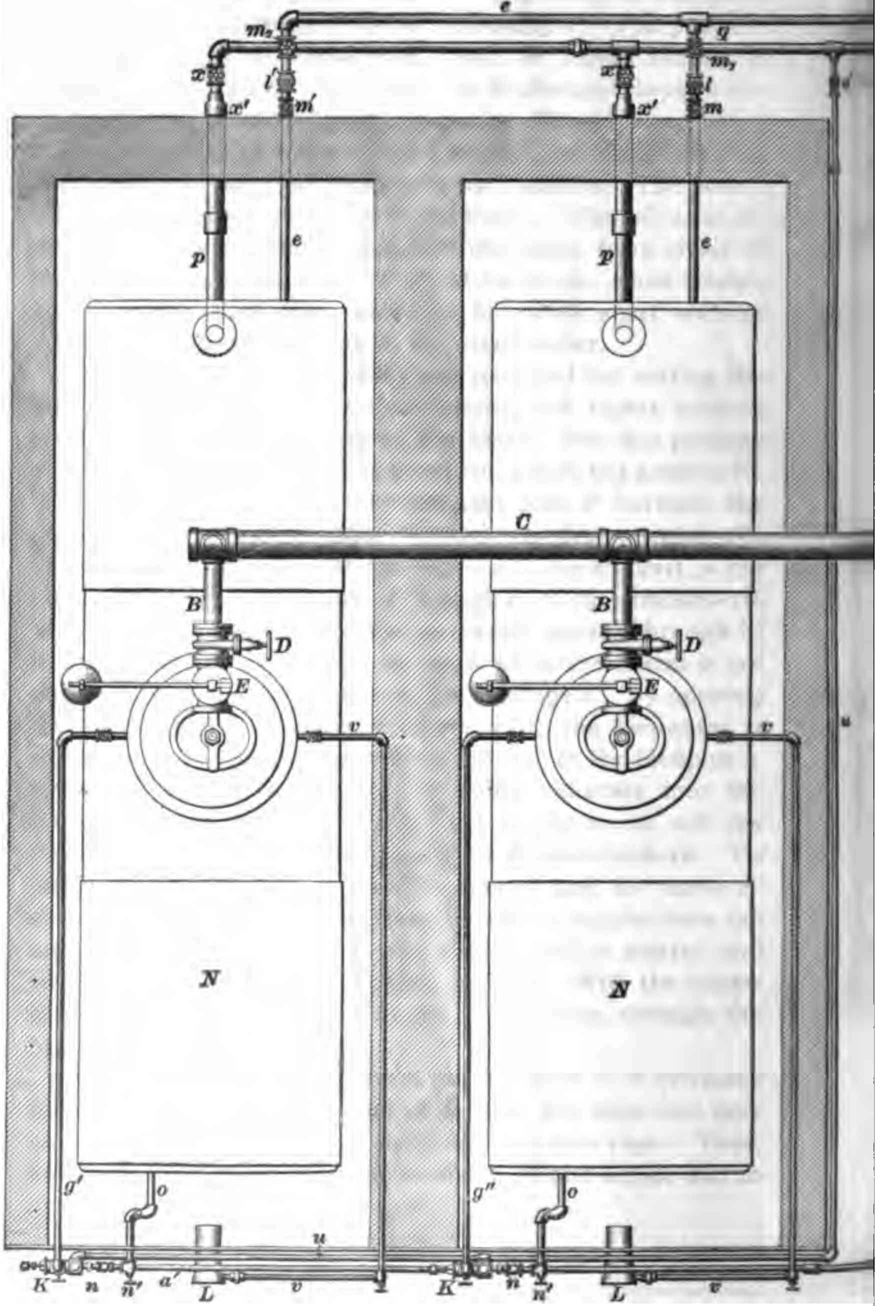
54. In order to illustrate the general arrangement of steam plants, particularly with reference to the arrangement of steam boilers, steam piping, and engines, a few typical cases are given. No rules that can be given are applicable to all cases in arranging the piping of a steam plant; each plant must be designed to suit the service for which it is intended and the local conditions. Figs. 34 and 35 will convey an idea of the way a plant and the piping may be arranged. Fig. 34 is an elevation and Fig. 35 a top view of the plant, which has two return-tubular boilers *N*, *N* so arranged that either boiler may be used at will or both may be run at the same time. In the following description, the letters refer to both figures:

Suppose that the boilers have been partly filled with water, and the fire started, and that the steam gauge *P* registers the desired pressure. The valves *D*, *D* are then opened and the steam is conveyed to the engine *M* through the short vertical pipes *A*, *A*, the short branch pipes *B*, *B*, and the main steam pipe *C*. It will be noticed that the safety valves *E*, *E* are attached to the upper ends of the pipes *A*, *A* and that the valves *D*, *D* are situated between the safety valves and the main steam pipe. Hence, there is no possibility of cutting off the connection between each boiler and its safety valve.

The water in the closed feedwater heater *H* is heated by the exhaust steam from the engine, which is conveyed to the heater by the exhaust pipe F. The water in the boiler is replenished by means of the feed-pump I. The pump is connected to a city reservoir, river, or other source of supply by the pipe a. The water is discharged through the delivery pipe d into the closed heater H, and after being heated is forced by the continued working of the pump into the boilers through the pipe e and its branches. The check-valves l, l' prevent the return of the water. The valves m, m' are for the purpose of shutting off the water from either of the boilers, if so desired. The valves m_1, m_2, when closed, permit either feed check-valve to be taken apart without interfering with the feeding of the other boiler.

Suitable by-pass connections are provided for cutting the heater out of service for examination and repair without interfering with the running of the plant. For this purpose a separate exhaust pipe F_1 is provided, which has a valve F'. A valve F'' is placed in the exhaust pipe F between the junction of F and F_1 and the heater. By first opening the valve F' and then closing the valve F'', the exhaust is cut off from the heater and passes through F_1 to the atmosphere. While the heater is in use, the feedwater passes through it; in order that the pump may be used while the heater is cut out, the by-pass pipe d' leads to the feedpipe e. By opening the valve d_1 and closing the valves d_2, d_3, the feedwater is cut off from the heater and passes directly to the feedpipe e. With the heater in service, the pump exhausts into the heater through the pipe c; when the heater is cut out, the pump exhausts through the pipe c_1 to the atmosphere. To make the change, the valve c' is opened and the valve c'' closed. The feed-pump receives its steam supply from the main steam pipe C through the pipe b, and is started and stopped by operating the throttle valve y. With the heater in use, the exhaust passes to the atmosphere through the pipe G.

While, in this case, a separate exhaust pipe F_1 is provided for use when the heater is out of service, the same end may be attained by connecting F and G by a by-pass pipe. Thus, an elbow may be placed directly above F' and a pipe run to



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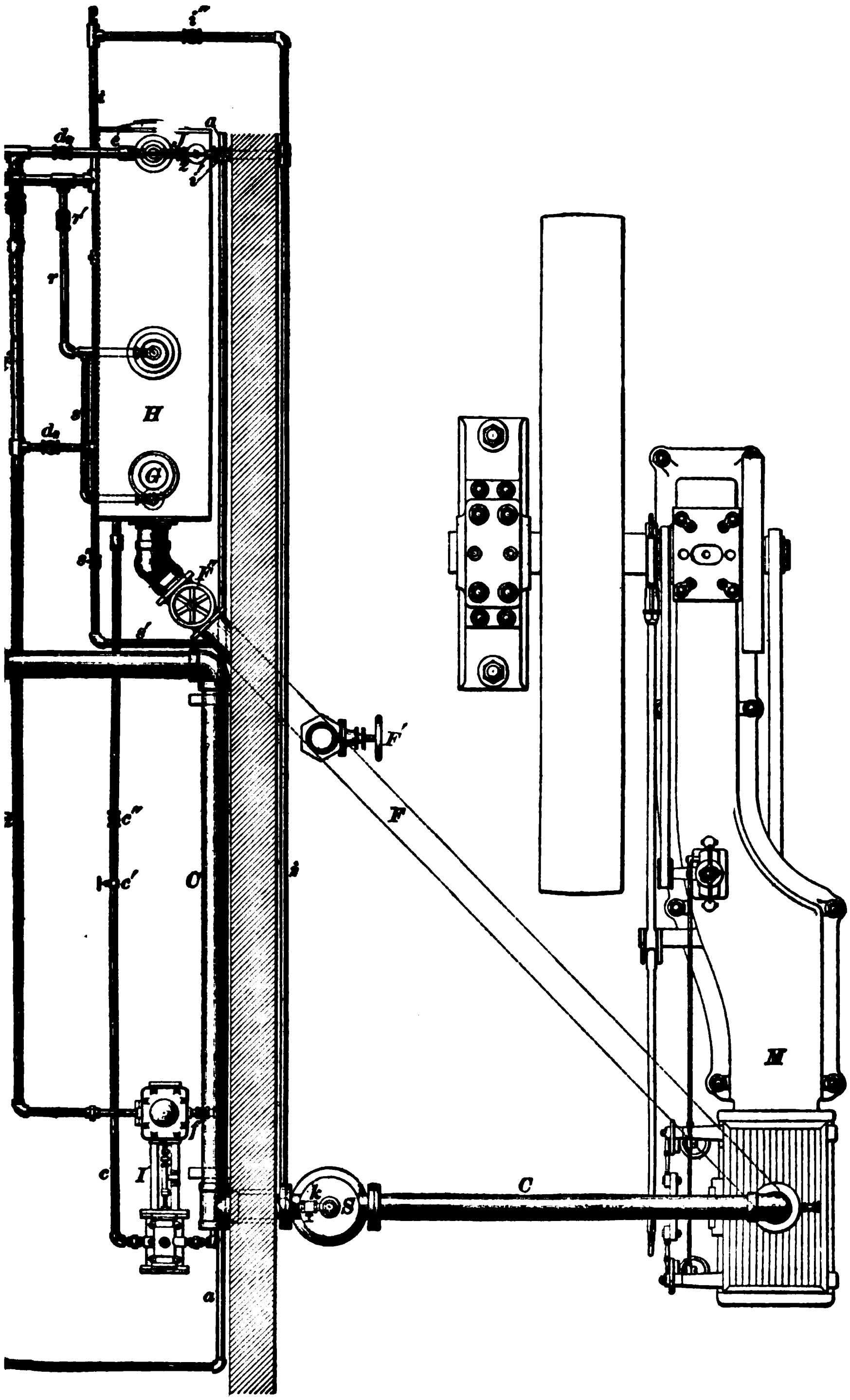
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a T placed in G; in that case, a valve will have to be placed in G below the T. This valve and the valve F'' will be closed and the valve F' opened in order to cut out the heater. A safety valve J is generally placed in the feedpipe to prevent any overpressure in the heater.

The blow-off pipes p, p are connected to the pipe q, which in turn is connected to the pipe t. The valves are shown at x, x. The pipes p are larger than q, for the reason that they are more likely to become choked with sediment than q. In many cases, the blow-off pipes p are independent; that is, they are not led into the same pipe. With the arrangement shown in the figure, there is no way of discovering a leak in the valves x, x should one occur. Should they become choked, they may be readily disconnected at x' from the pipe q and the sediment removed. The pipe r is the blow-off for the heater; it connects with the main blow-off pipe q.

Since the exhaust steam condenses more or less in passing through the heater, it is necessary to provide some means for getting rid of the condensed portion; this is accomplished by connecting the tubes to the blow-off pipe r by means of the pipe s. Valves are placed in r and s close to the heater; a valve r' is placed near the junction of r with q and is used for shutting off connection between the blow-off pipe and the heater drains when the heater is out of service. A small drain pipe s' is fitted to the exhaust pipe F, and connects with the blow-off pipe t. The globe valve s'' should be opened before the engine is started, so as to clear the exhaust pipe of water that may have accumulated in it. This valve should be closed again after the exhaust pipe is thoroughly warmed up and cleared of all water.

Steam-jet blowers L, L produce a forced draft, the steam required being obtained from the dome by the pipes v, v.

In case the pump is out of order or the heater disabled and it is considered advisable to feed hot water, the boilers can be fed by the injectors K, K. The steam for working the injectors is taken from the domes through the pipes g', g''. The water is led to the injectors by the pipe a', which is a continuation of the pump feedpipe a and delivers the water into the boiler through the feedpipes *o*, *o*. Before starting the injectors, the valve *f* should be closed so as to shut off the water supply from the pump. The pipe *u* conducts the overflow water from the injectors to the blow-off pipe *q*, and a valve *u'* is placed in the pipe *u*; this valve must be closed, when the boilers are being blown down, in order to prevent the water from backing into the overflow pipe. The check-valves *n*, *n* prevent the escape of water from the boiler through the injectors after they have stopped working. The globe valves *n'*, *n'* are additional safeguards; they are for the purpose of preventing the boilers from emptying themselves after a shutdown, in case an obstruction should prevent the check-valves from closing.

A separator *S* removes the entrained water from the steam. As will be seen, it is attached to the main steam pipe *C* and is also supported by the rod *h*, which is attached to the beam overhead. The water thus removed flows down by gravity through the pipe *i* into the feedpipe *e* just below the safety valve *J* on the feedwater heater. As the temperature of the water is fully 212° F., it is not necessary that it should pass through the heater. When the separator discharges directly into the boiler feedpipe, the bottom of the separator should be at least 2 feet above the highest water level in the boiler, since, if both were at the same level, the pump would force water into the separator and thus destroy its action. The difference in levels between the bottom of the separator and the water level in the boiler is the head that causes the flow. The pipe *i* is fitted with the globe valve *k* and the check-valve *z*. The drain pipe *i* of the separator is also connected to the waste pipe *t* in order to allow the separator to be drained whenever the feedpipe *e* is out of service. To drain into *t*, the valve *i'* is closed and *i''* opened.

55. Fig. 36 shows two views, a plan and an elevation, of the arrangement of a small electric-light station in which the steam piping is of the simplest character. The station contains two 100-horsepower boilers and one 300-horsepower boiler, all of the water-tube type, which supply steam to two

tandem-compound engines. The one engine has cylinders 13 inches and 23 inches in diameter, and a stroke of 15 inches; the other engine has cylinders 17 inches and 29 inches in diameter, and a stroke of 18 inches. Both engines are direct-connected to dynamos.

The branch steam pipes from the three boilers deliver into a horizontal steam main placed at the level of the drums, an angle stop-valve being placed over each boiler, and each branch connecting to the top of the main with a long-radius bend, as shown. The supply pipe for each engine is taken from the top of the main, each supply pipe being provided with an angle stop-valve and a throttle valve placed close to the engine. A steam separator is placed in each engine supply pipe directly over the throttle valve. Owing to the method in which the piping is run from the boilers to the engines, it is quite flexible, so that there will be little or no stress set up by its expansion or contraction.

The exhaust piping is shown by dotted lines in the plan view. The exhaust pipes from the two engines are placed below the floor and are joined by means of a Y fitting to the main exhaust pipe, which conveys the exhaust steam through a closed feedwater heater, provided with a by-pass, to the atmosphere. A separator intended to remove oil, etc. from the exhaust steam before it reaches the heater is placed in the main exhaust pipe where shown.

The various pipe lines used for draining piping as well as the heaters and the separators, etc., the boiler feed-system, the fire-service pipes, the boiler blow-offs, and similar small piping found in a steam plant, are not shown in Fig. 36, the purpose of this illustration being chiefly to show the arrangement of the main pipes.

FUELS AND BOILER TRIALS

FUELS

COMBUSTION OF FUELS

THEORY OF COMBUSTION

1. Since **fuel,** chiefly in the form of coal, is the source of all energy made available by the steam engine, a study of the principles governing the economical generation of power by means of the steam engine must begin with a study of the combustion of fuel and of means of preventing a waste of heat, by guarding against those conditions that tend toward incomplete combustion. Heat lost by imperfect combustion cannot be recovered.

2. Definition.—Combustion may be defined as a rapid chemical combination of two or more substances producing heat. The ordinary combustion that takes place in a furnace is the chemical combination of the carbon and hydrogen, of which the fuel is composed, with the oxygen of the air. This chemical combination produces intense heat, which may be used for generating steam or for any other purpose. Carbon and oxygen, or hydrogen and oxygen, will not combine at ordinary temperatures; their temperature must first be raised to a fixed temperature, called the igniting temperature, before the attraction between the two is sufficient to cause them to combine. As soon, however, as combustion is once started, the temperature is kept up to the igniting point, or above it, by the combustion itself, and as long as the oxygen and carbon or hydrogen are supplied combustion will continue.

3. Combustible Substances in Coal.—The combination of oxygen with the carbon and hydrogen of the fuel produces gases, which pass away through the flues and chimney. Bituminous coal is composed largely of carbon and hydrogen. When the coal is heated, compounds of hydrogen and carbon, called **hydrocarbons,** are driven off, partly in the form of permanent gases, and partly as vapors that may easily be condensed or changed to a liquid form. The process of separating the gases and vapors from the part of the coal that cannot be vaporized by the mere action of heat is called **distillation;** the substances driven off from the coal by heat are called **volatile substances,** while the portion remaining is called **coke,** which is composed chiefly of carbon. This carbon is called the **fixed carbon** of the coal.

4. The volatile substances may be divided into two classes: **non-combustible** and **combustible** substances. The first class consists mostly of water, free oxygen, and nitrogen; these are driven off when the coal is heated, the water as steam and the gases in their free state. The second class consists of the hydrocarbons, which comprise numerous compounds of hydrogen and carbon. When the coal is heated, part of the hydrocarbons are driven off in a gaseous form and part as vapors.

5. The principal gases in the volatile combustible are **carbureted hydrogen,** or **marsh gas,** consisting of one atom of carbon and four atoms of hydrogen, as shown by its symbol CH_4, and **olefiant gas,** C_2H_4. With many coals, free hydrogen is given off in considerable quantities; small quantities of other less important gases are also generally present.

The vapors are mostly coal tar and naphtha, with small quantities of sulphur. Their presence can be detected by placing a cold iron bar into the yellow gases rising from a fresh charge; a sticky coating, consisting mostly of the condensed tar, will form on the cold metal.

6. The proportion of the volatile matter in coal depends on its composition. Anthracite consists almost entirely of fixed carbon and ash; in some bituminous coals, the greater part is volatile. The relative proportions of fixed gases and condensable vapors in the volatile parts of coal also vary with the composition of coal. In some cases, the volatile matter contains considerable quantities of the tarry vapors and heavy hydrocarbon gases, as C_2H_4, while in others it consists largely of the light marsh gas CH_4 and free hydrogen.

The quantity and composition of the volatile matter depend not only on the composition of the coal itself, but also on the conditions under which distillation takes place, differences in temperature and the presence or absence of air or steam modifying the composition of the vapor and gases to a great extent. Irregularities in firing and draft result in great differences in the quantity and composition of the gas burned in the furnace.

7. With few exceptions, coal contains small quantities of sulphur, usually in combination with some other element; one of the most common compounds is that of sulphur and iron, known as **iron pyrites**. When the coal is heated, the sulphur is separated from the iron and burns to sulphur dioxide, SO_2. The heat derived from the combustion of the sulphur found in the coal is small, but the sulphur dioxide formed, in combination with the moisture in the gases, corrodes iron very rapidly; any relatively cold metal exposed to gases from coal rich in sulphur is rapidly corroded and destroyed.

8. Igniting Temperature of Carbon and Volatile Substances.—At the temperature generally existing in a boiler furnace, the tar and other liquids vaporize and mix with the gases. This gaseous mixture is readily burned under proper conditions of air supply and temperature. In order to burn the fixed carbon in the coal, a high temperature is needed to cause the atoms of carbon to combine with the oxygen supplied by the air. The igniting temperature of the fixed carbon and also of the volatile substances is estimated to be about 1,800° F. Since the maximum temperature in the furnace rarely exceeds 2,500° F. and is ordinarily several hundred degrees less, it is seen that on account of the relatively small difference between the igniting temperatures of the carbon and gases and the maximum temperature in the furnace, constant care is needed to prevent the temperature in the furnace from falling below 1,800° F.

When the supply of air is sufficient and the temperature high enough, the carbon burns to carbon dioxide, CO_2, which, being the product of complete combustion, is incombustible. With a high temperature and deficient air supply, carbon monoxide, CO, is formed. Since this gas is the product of incomplete combustion, it can be burned to CO_2 by bringing it into contact with air while highly heated.

9. Chemical Reactions.—When solid carbon burns on a grate, the chemical changes, or reactions, are about as follows: Some of the oxygen of the air that rises through the grate combines with the first layers of hot carbon in the proportion of two atoms of oxygen to one atom of carbon, forming carbon dioxide, CO_2. As the gases rise through the fire, more of the oxygen combines with carbon, and as long as the supply of air is sufficient and well distributed, the combination is mostly in the proportion that produces CO_2. With a thick bed of fuel, however, or an arrangement of the fuel that does not permit a proper distribution of the air, there will be some portions of the fire in which the supply of oxygen is not great enough to furnish the two atoms for the production of CO_2; only one atom of oxygen will be available for combination with some of the carbon atoms burned, and therefore the product will be carbon monoxide, CO. Further, when a molecule of CO_2 comes into close contact with the hot carbon, the attraction of the carbon for oxygen is so great that one atom of the oxygen leaves the CO_2 and combines with an atom of carbon; two molecules of CO are thus formed, one by the separation of one of the oxygen atoms from the molecules of CO_2 and the other by the combination with an atom of carbon of the oxygen atom so released. The separation of a given weight of CO_2 into CO and O absorbs as much heat as was developed when the CO combined with O to form CO_2. The net production of heat is the same whether a certain amount of carbon is burned to CO directly and passes off in that form, or a part of it is first burned to CO_2, this gas being then decomposed with the production of CO and O, the latter combining with the remainder of the carbon to form CO. The carbon monoxide formed in the fuel bed passes into the furnace, and if there is not sufficient oxygen present or if the temperature is not high enough, it will pass away unburned. If sufficient oxygen is present and the furnace temperature is high enough, each molecule of carbon monoxide will combine with another atom of oxygen and thus burn to carbon dioxide.

10. A careful study of the foregoing outlines of the processes involved shows that economical combustion, both of the volatile matter and of the solid carbon, involves the following essential conditions: (1) There must be a supply of air sufficient to furnish the oxygen required for complete combustion. (2) This air must be so distributed as to bring the oxygen into contact with all parts of the fuel. (3) The temperature must be high enough to bring about the combustion. With any of these essentials lacking, there will be incomplete combustion and loss of heat.

AIR SUPPLY FOR COMBUSTION

11. Weight of Air Required.—The oxygen required for combustion is taken from the air. Air is composed of twenty-three parts (by weight) of oxygen and seventy-seven parts of nitrogen. More exactly, the percentage of oxygen is 23.185 and of nitrogen 76.815; but for practical purposes, and for the solution of the following problems, 23 and 77 may be taken. The nitrogen takes no part in the combustion.

and simply passes off through the chimney, carrying a certain quantity of heat with it.

12. Carbon dioxide, CO_2, the product of complete combustion, is composed (by weight) of twelve parts of carbon and thirty-two parts of oxygen. Hence, to burn 1 pound of carbon requires $32 \div 12 = 2\frac{2}{3}$ pounds of oxygen. If the oxygen is taken from the air, it will take $2\frac{2}{3} \div .23 = 11.6$ pounds of air to supply the $2\frac{2}{3}$ pounds of oxygen. This is because only 23 per cent. of air is oxygen. The combustion of 1 pound of carbon may be represented as follows:

	Elements	Products
1.0 pound carbon .	1.00 pound carbon . .	3.67 pounds carbon dioxide
11.6 pounds air . . .	2.67 pounds oxygen . .	
	8.93 pounds nitrogen .	8.93 pounds nitrogen
12.6	12.60	12.60

That is, 1 pound of carbon requires 11.6 pounds of air for complete combustion. Of this air, 2.67 pounds is oxygen, which combines with the pound of carbon, forming 3.67 pounds of carbon dioxide. The 8.93 pounds of nitrogen contained in the air pass off with the products of combustion.

Take, next, the complete combustion of 1 pound of hydrogen. The product of the combustion is water, H_2O, which is composed by weight of two parts hydrogen to sixteen parts oxygen. Hence, 1 pound of hydrogen requires $16 \div 2 = 8$ pounds of oxygen to unite with it. The air required to furnish 8 pounds of oxygen is $8 \div .23 = 34.8$ pounds. The process of combustion, therefore, is as follows:

	Elements	Products
1.0 pound hydrogen	1.0 pound hydrogen	9.0 pounds water
34.8 pounds air . .	8.0 pounds oxygen .	
	26.8 pounds nitrogen	26.8 pounds nitrogen
35.8	35.8	35.8

13. There is one other case that may occur: the combustion of carbon may not be complete. If insufficient air or oxygen is supplied to the burning carbon, it is possible for the carbon and oxygen to form another gas, carbon monoxide, CO, instead of carbon dioxide, CO_2.

The combustion of 1 pound of carbon to form carbon monoxide requires only one-half the oxygen that would be necessary to form carbon dioxide. This is because in carbon monoxide one atom of carbon unites with one atom of oxygen instead of two. To burn 1 pound of carbon to carbon dioxide requires 11.6 pounds of air; therefore, to burn it to carbon monoxide will require but 5.8 pounds of air.

14. The quantities of air required for combustion are shown in the following tabulation:

1 Pound	Air at 62°	Products of Combustion
Hydrogen	34.8 lb., or 457 cu. ft.	Water Nitrogen
Carbon burned to carbon dioxide	11.6 lb., or 152 cu. ft.	Carbon dioxide Nitrogen
Carbon burned to carbon monoxide	5.8 lb., or 76 cu. ft.	Carbon monoxide Nitrogen

15. The **minimum quantity of air** required for the complete combustion of any given fuel may readily be found when its chemical composition is known. It has been shown that 1 pound of carbon requires 11.6 pounds of air, and 1 pound of hydrogen requires 34.8 pounds of air for complete combustion. Letting C represent the percentage of carbon and H the percentage of hydrogen contained in fuel, the minimum quantity of air required for the complete combustion of a pound of fuel must be $11.6\,C + 34.8\,H$. If the fuel contains oxygen, the oxygen will unite with one-eighth its weight of hydrogen to form water, and the weight W of air required will be

$$W = 11.6\,C + 34.8\left(H - \frac{O}{8}\right)$$

where O represents the percentage of oxygen in the fuel.

Example.—A certain kind of coal has the following chemical composition: Carbon, .80; hydrogen, .08; oxygen, .12. Find the minimum quantity of air required for complete combustion of 1 pound of coal.

Solution.—Using the formula,

$$W = 11.6\,C + 34.8\left(H - \frac{O}{8}\right) = 11.6 \times .8 + 34.8\left(.08 - \frac{.12}{8}\right) = 11.54 \text{ lb.}$$

Ans.

The minimum volume of air, in cubic feet, may be found by multiplying the result obtained from the preceding formula by 13.14, which is the volume, in cubic feet, of 1 pound of air at 62° F. Owing to the difficulty of perfectly mixing the air with the thick bed of burning fuel, complete combustion cannot be obtained in practice by using the theoretical quantity of air given by this formula.

Furnaces with ordinary chimney draft require about double, and furnaces with forced draft about one and one-half times the theoretical quantity of air.

TEMPERATURE, HEAT, AND RATE OF COMBUSTION

16. Heat of Combustion.—It has been determined, by direct experiment, that 1 pound of carbon burned to carbon dioxide gives out 14,600 British thermal units of heat; burned to carbon monoxide it gives out only 4,400 British thermal units. The complete combustion of 1 pound of hydrogen gives out about 62,000 British thermal units. The heat of combustion of a given fuel may be determined approximately by the following formula:

$$h = 14{,}600\,C + 62{,}000\left(H - \frac{O}{8}\right)$$

where h is the heat of combustion in British thermal units and C, H, and O have the same meaning as in the formula of Art. **15**.

EXAMPLE.—The composition of a variety of coal is as follows: Carbon, 82 per cent.; hydrogen, 4 per cent.; oxygen, 6 per cent.; other substances, 8 per cent. Find the approximate amount of heat developed by the combustion of 1 pound of this coal.

SOLUTION.—Using the formula,

$$h = 14{,}600\,C + 62{,}000\left(H - \frac{O}{8}\right) = 14{,}600 \times .82 + 62{,}000\left(.04 - \frac{.06}{8}\right)$$
$$= 13{,}987 \text{ B. T. U. Ans.}$$

17. The formula of Art. **16** gives only approximate results, and where the heating power of a fuel must be known accurately, as, for example, in evaporative tests of steam boilers, it is better to determine experimentally the heat of combustion by means of a coal calorimeter.

The total heat required to raise 1 pound of water at 62° F. to 212° F. and completely evaporate it at the latter temperature is, from the Steam Tables, 1,116.6 British thermal units, and the heat required to evaporate 1 pound of water at 212° is 965.8 British thermal units. Consequently, 1 pound of carbon completely burned to carbon dioxide should raise $14,600 \div 1,116.6 = 13$ pounds of water from 62° to 212° and evaporate it at the latter point, or it should vaporize $14,600 \div 965.8 = 15$ pounds of water from and at 212° F. In a similar manner, the evaporative power of any fuel may be obtained by dividing its heat of combustion by 1,116.6 for an evaporation from 62° or by 965.8 for an evaporation from 212°.

In actual practice, the above theoretical values of the evaporative powers of fuels cannot be attained on account of the loss of heat by radiation and other causes. The best boilers can evaporate about 11 to 12½ pounds of water from and at 212° per pound of combustible.

18. Temperature of Combustion.—The temperature of the fire depends on the total heat of combustion and on the products of combustion. The specific heats of the products of combustion are as follows:

Air	.23751	Carbon dioxide .	.2170
Oxygen . . .	.21751	Nitrogen	.2438
Steam	.4805	Carbon monoxide	.2479

In the case of the complete combustion of carbon, it has been found that per pound of carbon there was given off 3.67 pounds of carbon dioxide and 8.93 pounds of nitrogen; the heat of combustion was found to be 14,600 British thermal units per pound of carbon.

The heat of combustion is given up to the products of combustion, the carbon dioxide and nitrogen. The heat required to raise these products 1° in temperature is

$$3.67 \times .2170 + 8.93 \times .2438 = 2.9735 \text{ British thermal units}$$

Hence, the total rise in temperature above atmosphere is

$$14,600 \div 2.9735 = 4,910° \text{ F.}$$

To take a more complicated case, suppose that the fuel is coal containing the elements in the following proportions:

Carbon .84, hydrogen .06, and oxygen .10. Required, to find the temperature of the furnace, supposing that double the theoretical quantity of air is mixed with the fuel.

Making allowance for the water due to the combination of H and O in the fuel, the free hydrogen is

$$H - \frac{O}{8} = .06 - \frac{.10}{8} = .0475$$

and the composition of the coal may be written, carbon .84, hydrogen .0475, and water .1125, since the combined weight of hydrogen and oxygen is $.10 + .06 = .16$, and subtracting from this the weight of free hydrogen, the remainder, which is water, is $.16 - .0475 = .1125$. The heat of combustion per pound of coal is, therefore, by the formula given in Art. **16**, $.84 \times 14,600 + .0475 \times 62,000 = 15,209$ British thermal units.

The minimum air required is, by the formula given in Art. **15**,

$$11.6 \times .84 + 34.8 \times .0475 = 11.4 \text{ pounds, nearly;}$$

the air furnished is therefore $11.4 \times 2 = 22.8$ pounds.

The products of the combustion are

Carbon dioxide	$.84 \times 3.67 =$	3.08 lb.
Water (steam) . . .	$.1125 + .0475 \times 9 =$	.54 lb.
Nitrogen	$11.4 \times .77 =$	8.78 lb.
Free air		11.4 lb.
		23.8 lb.

The extra 11.4 pounds of air simply passes up the chimney with the other products of combustion. The object of supplying it was to insure perfect combustion.

From the total heat of combustion must be subtracted the heat required to change the water into steam, that is, $.54 \times 965.8$ British thermal units. The resulting temperature is

$$t = \frac{15,209 - .54 \times 965.8}{3.08 \times .217 + .54 \times .4805 + 8.78 \times .2438 + 11.4 \times .23751} = 2,542.9° \text{ F.}$$

above the temperature of the atmosphere.

If the theoretical quantity of air had been supplied instead of double that amount, the temperature would have been about 4,800° F.; hence, it is apparent that air in excess of the minimum amount required for combustion dilutes the products of combustion and lowers the temperature of the furnace. It is very important that just the proper amount of air be supplied to insure complete combustion; a greater quantity leads to loss of heat by diluting the products of combustion, while on the other hand an insufficient quantity leads to loss through incomplete combustion.

19. Rate of Combustion.—The rate of combustion of the fuel in the furnace is usually stated in pounds per hour burned on each square foot of grate. For coal, the usual rates of combustion under natural draft are as follows:

	Pounds per Square Foot per Hour
Slowest rate of combustion in Cornish boilers	4 to 6
Ordinary rate of combustion in Cornish boilers	10 to 15
Ordinary rates in factory boilers	12 to 18
Ordinary rates in marine boilers	15 to 25
Quickest rates of complete combustion of anthracite, the air being supplied through the grate only	15 to 20
Quickest rates of complete combustion of bituminous coal, with air holes above the fuel one thirty-sixth area of grate	20 to 25
Under forced draft, the usual rates of combustion are as indicated below:	
Locomotives	40 to 100
Torpedo boats	60 to 125

TRANSFER AND LOSS OF HEAT

20. The **transfer of heat** from the burning fuel on the grate of the furnace to the water of the boiler is accomplished by radiation, conduction, and convection. It is estimated that when the fire is burning brightly, about one-half of the heat received from the furnace by the boiler is radiated. The transfer of heat through the water is due to convection, since liquids are poor conductors of heat. The particles of water next to the shell become heated and immediately rise into the main body of water, giving place to fresh particles of cold water. The rapidity with which heat will be absorbed by convection, therefore, depends on the effectiveness of the water circulation in the boiler and on the extent and conductivity of the heating surfaces. The transference of heat through the shell and furnace plates takes place by conduction. It has been shown experimentally that the quality or thickness of the material has little influence, thick iron tubes working practically as well as thin brass ones. Very thick plates, however, are liable to be injured by burning when exposed to the direct action of the fire.

21. The rapidity of the transfer of heat by convection depends on the rapidity of circulation; besides this, the circulation is useful in preventing, to a certain extent, the accumulation of deposits of sediment from the feedwater. Again, a rapid circulation keeps the parts of the boiler at a uniform temperature and is a safeguard against overheating.

Fig. 1 shows the effect of the circulation on the transfer of heat from the products of combustion in the furnace to the

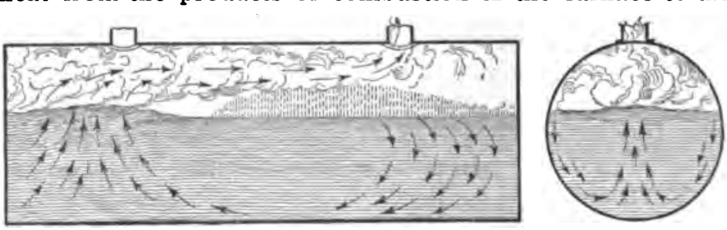


FIG. 1

water in a plain cylindrical boiler. The heated currents of water rise from the hottest part of the shell directly over the furnace and carry bubbles of steam to the surface. The cooler water rushes in to take the place of the ascending water, and thus the circulation is maintained. As shown in the figure, there are two currents, one carrying the cold water from rear to front, and the other carrying it down the outside of the shell and up through the center. The circulation is in a direction opposite to that of the furnace gases.

Since, in all cylindrical shells, the water is contained in a solid mass, broken only by flues or tubes, the circulation is more or less interfered with by opposing currents. The circulation is more rapid and effective if the water is constrained to follow a particular path. This is one of the strong points of the water-tube boiler. The water must pass in one direction through a series of tubes; hence, the circulation is rapid and uninterrupted. The difference between the cylindrical and the water-tube boiler in this respect may be illustrated as follows: The cylindrical boiler, with its contained mass of water, may be compared to an ordinary kettle

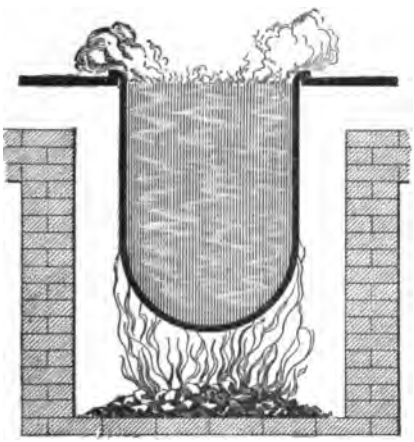


FIG. 2

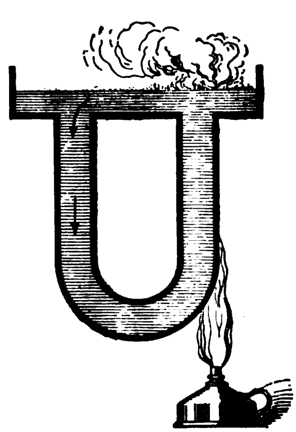


FIG. 3

in the process of boiling. (See Fig. 2.) The water rises rapidly around the outer edges and flows downwards in the center. If, however, the fire is quickened, the upward and downward currents interfere and the kettle boils over. The water-tube boiler is identical in principle with a **U** tube suspended from a vessel of water (Fig. 3), with the heat applied to one leg. The circulation is set up immediately and proceeds quietly, no matter how hot the fire may be.

22. Loss of Heat.—A portion of the heat generated by the burning fuel in the furnace is usefully expended in evaporating water. There is, however, more or less loss of heat due to the following causes:

1. A certain amount of heat is carried up the chimney by the waste gases, the temperature of which is generally between 400° and 600°.

2. In some cases, heat is lost by incomplete combustion.

3. Heat is required to vaporize the water formed by the combustion of hydrogen.

4. The escape of free carbon in the form of smoke is a loss.

5. Heat is lost by radiation.

The loss due to the first of these causes cannot be avoided where chimney draft is used, since the temperature of the ascending gases must be from 300° to 600° F. to insure a good draft. The loss may, however, be aggravated by using an excess of air.

The loss due to the second cause may be avoided by using a sufficient quantity of air to insure complete combustion.

The loss of the heat required to vaporize the water in the fuel is unavoidable; with fuels that have been wetted, it is, of course, still greater.

The formation of smoke is a very common and very fruitful source of waste. Bituminous coals rich in hydrocarbon gases produce the most smoke. The volatile hydrocarbons are driven off by the high temperature, and come in contact with the air; the oxygen of the air unites with the more inflammable hydrogen, leaving the carbon in a finely divided condition; on being cooled, the fine particles of carbon appear as smoke.

Smoke may be avoided by bringing a supply of fresh air in contact with the carbon while it is red hot in the flame, that is, before it cools and becomes smoke. A successful method of smoke prevention consists in forcing air into the furnace above the fuel by means of steam jets.

The loss of heat by radiation may be made small by covering exposed parts of the boiler with non-conducting materials. The radiation from an internally fired boiler is less than from one externally fired, the excess of the latter over the former being due to the brickwork furnace walls.

EXAMPLES FOR PRACTICE

1. A certain coal has the following chemical composition: Carbon, 90.4 per cent.; hydrogen, 3.3 per cent.; oxygen, 3.0 per cent.; other substances, 3.3 per cent. Calculate the heat of combustion of 1 pound.
Ans. 15,012 B. T. U.

2. The chemical composition of marsh gas is, carbon, 85.7 per cent.; hydrogen, 14.3 per cent. (*a*) Find the heat of combustion of 1 pound. (*b*) Find the weight of air per pound necessary for complete combustion.
Ans. $\begin{cases} (a) \text{ 21,378 B. T. U.} \\ (b) \text{ 14.92 lb.} \end{cases}$

3. The chemical composition of a certain kind of peat is as follows: Carbon, 60 per cent.; hydrogen, 6 per cent.; oxygen, 31 per cent.; ash, 3 per cent. (*a*) Find the heat of combustion of 1 pound of peat. (*b*) Find the weight of air per pound of peat required for complete combustion.
Ans. $\begin{cases} (a) \text{ 10,078.2 B. T. U.} \\ (b) \text{ 7.7 lb.} \end{cases}$

FUELS USED IN STEAM MAKING

23. The fuels used in the generation of steam are chiefly coal, coke, wood, the mineral oils (as petroleum), and natural gas. Other fuels, such as the waste gases from blast furnaces, straw, bagasse (refuse from sugar cane), dried tan bark, green slabs, sawdust, peat, are also used. All these fuels are composed either of carbon alone or carbon in combination with hydrogen, oxygen, and non-combustible substances.

CLASSIFICATION OF COAL

24. Leading Varieties.—A well-known authority, William Kent, divides coal into four leading varieties, as follows:

1. *Anthracite*, which contains from 92.31 to 100 per cent. of fixed carbon, the remainder being volatile hydrocarbons.
2. *Semianthracite*, which contains from 87.5 to 92.31 per cent. of fixed carbon, the remainder being volatile hydrocarbons.
3. *Semibituminous coal*, which contains from 75 to 87.5 per cent. of fixed carbon, the remainder being volatile hydrocarbons.

4. *Bituminous coal*, which contains from 0 to 75 per cent. of fixed carbon, the remainder being volatile hydrocarbons.

25. Anthracite is comparatively difficult to ignite and requires a strong draft to burn it. It is quite hard and shiny; in color it is a grayish black. It burns with almost no smoke; this fact gives it a peculiar value in places where smoke is objectionable.

Anthracite is known in commerce by different names, according to the size into which the lumps are broken. These names, with the generally accepted dimensions of the screens over and through which the lumps of coal will pass, are given below, the meshes all being square:

Culm passes through $\frac{5}{64}$-inch mesh.

No. 3 Buckwheat passes over $\frac{5}{64}$-inch mesh and through $\frac{1}{8}$-inch mesh.

No. 2 Buckwheat passes over $\frac{1}{8}$-inch mesh and through $\frac{1}{4}$-inch mesh.

No. 1 Buckwheat passes over $\frac{1}{4}$-inch mesh and through $\frac{1}{2}$-inch mesh.

Pea passes over $\frac{1}{2}$-inch mesh and through $\frac{3}{4}$-inch mesh.

Chestnut passes over $\frac{3}{4}$-inch mesh and through $1\frac{3}{8}$-inch mesh.

Stove passes over $1\frac{3}{8}$-inch mesh and through 2-inch mesh.

Egg passes over 2-inch mesh and through $2\frac{3}{4}$-inch mesh.

Broken, or *grate*, passes over $2\frac{3}{4}$-inch mesh and through $4\frac{1}{2}$-inch bars.

Steamboat passes over bars $4\frac{1}{2}$ inches apart and through bars 6 inches apart.

Lump passes over bars set 6 inches apart.

26. Semianthracite kindles easily and burns more freely than the true anthracite; hence, it is highly esteemed as a fuel. It crumbles readily and may be distinguished from anthracite by the fact that when just fractured it will soil the hand, while anthracite will not do so. It burns with very little smoke. Semianthracite is broken into different sizes for the market; these sizes are the same and are known by the same trade names as the corresponding sizes of anthracite.

27. Semibituminous coal differs from semianthracite only in having a smaller percentage of fixed carbon and more volatile hydrocarbons. Its physical properties are practically the same, and since it burns without the smoke and soot emitted by bituminous coal, it is a valuable steam fuel.

28. Bituminous coal may be broadly divided into three general classes:

1. *Caking Coal.*—This name is given to coals that, when burned in the furnace, swell and fuse together, forming a spongy mass that may cover the whole surface of the grate. These coals are difficult to burn, since the fusing prevents the air from passing freely through the bed of burning fuel; when caking coals are burned, the spongy mass must be frequently broken up with the slice bar, in order to admit the air needed for its combustion.

2. *Free-Burning Coal.*—This is often called non-caking coal from the fact that it has no tendency to fuse together when burned in a furnace.

3. *Cannel Coal.*—This is a grade of bituminous coal that is very rich in hydrocarbons. Its large percentage of volatile matter makes it valuable for gas making, but it is little used for the generation of steam, except near the places where it is mined.

29. Bituminous and semibituminous coals are known to the trade by the following names:

Lump coal, which includes all coal passing over screen bars $1\frac{1}{2}$ inches apart.

Nut coal, which passes over bars $\frac{3}{4}$ inch apart and through bars $1\frac{1}{2}$ inches apart.

Pea coal, which passes over bars $\frac{3}{8}$ inch apart and through bars $\frac{3}{4}$ inch apart.

Slack, which includes all coal passing through bars $\frac{3}{8}$ inch apart.

30. Lignite comes under the general head of bituminous coal. Properly speaking, it occupies a position between peat and bituminous coal, being probably of a later origin than the latter. It has an uneven fracture and a dull luster. Its value as a steam fuel is limited, since it will easily break in transportation. Exposure to the weather causes it to absorb moisture rapidly; it will then crumble quite readily. It is non-caking and yields but a moderate heat, and is in this respect inferior to even the poorer grades of bituminous coal.

31. Coke is made from bituminous coal by driving off its volatile constituents. It is used chiefly for metallurgical purposes, though it is a valuable fuel for steam purposes.

32. Wood is much used in localities where it is abundant. The effective heating values per pound of different kinds of wood differ but very little.

33. Bagasse is the refuse left after the juice has been extracted from the sugar cane by means of the mill rolls. Its use is limited to tropical and semitropical countries, where the sugar cane is grown.

Dried tan bark, straw, slabs, and sawdust being refuse, their use is local and usually confined to tanneries, threshing outfits, sawmills, and planing mills.

34. Petroleum is occasionally used as a fuel and possesses some advantages, among which are the ease of lighting and controlling the fire, uniformity of combustion, and economy in labor. Its disadvantages are: danger of explosion, loss by evaporation, and high cost in comparison with coal. The Standard Oil Company estimates that 173 gallons of petroleum is equal to 1 long ton (2,240 pounds) of coal, allowing for all savings incidental to its use.

35. Natural gas is abundant in parts of Ohio and Pennsylvania, and is there often used as a fuel for the generation of steam. On an average, 30,000 cubic feet of natural gas is the equivalent of 1 ton of coal.

36. Waste gases from the furnaces of rolling mills and from blast furnaces are extensively used. Naturally, their use is limited to the places where they are produced.

37. Peat may be classified as occupying an intermediate position between wood and coal. When first cut, it is totally unfit for fuel, since it contains from 75 to 80 per cent. of water. When dried, it makes a fairly good fuel. Generally speaking, 1 pound of good bituminous coal may be considered as equivalent to 2 pounds of dried peat, $2\frac{1}{2}$ pounds of dry wood, $2\frac{1}{2}$ to 3 pounds of dry tan bark or sun-dried bagasse, 3 pounds of cotton stalks, $3\frac{3}{4}$ pounds of straw, 6 pounds of wet bagasse, and from 6 to 8 pounds of wet tan bark.

STEAM-BOILER TRIALS

PURPOSE OF BOILER TRIALS

38. A **boiler trial,** or **boiler test** as it is often called, may be made for one or more of several purposes, the method of conducting the trial depending largely on its purpose. The boiler trial may vary from the simplest one, in which the only observations are the fuel burned and the water fed to the boiler in a stated period of time, to the elaborate standard boiler trial, in which special apparatus and several skilled observers are essential.

The object of a boiler trial may be to determine: the efficiency of the boiler under given conditions; the comparative value of different boilers working under the same conditions; the comparative value of fuel; the evaporative power, or the horsepower, of the boiler.

OBSERVATIONS AND MEASUREMENTS

39. The essential operations of a boiler trial are the weighing of the feedwater and fuel, and the observations of the steam pressure, temperature of feedwater, and various other less important pressures and temperatures. In conducting a boiler trial, the various observations of temperatures, pressures, etc. should be made simultaneously at intervals of about 15 minutes.

40. The coal supplied to the furnace is weighed out in lots of 500 or 600 pounds. It is a convenient plan to have a box with one side open placed on a platform scale. A weight is then placed on the scale beam sufficient to balance the box. The scale may then be set at 500 or 600 pounds, the coal shoveled in until the beam rises, and then fed directly from the box to the furnace.

After the test, the ashes and clinkers must be raked from the ash-pit and grate and weighed. This weight subtracted from the weight of the coal used gives the amount of combustible.

41. The amount of water evaporated in a test for comparative fuel values may be taken as equal to the amount of feedwater supplied without introducing any serious error. The most reliable method of measuring the feedwater delivered to the boilers is to weigh it. A convenient way of doing this is to have two tanks *a* and *b*, Fig. 4, one above

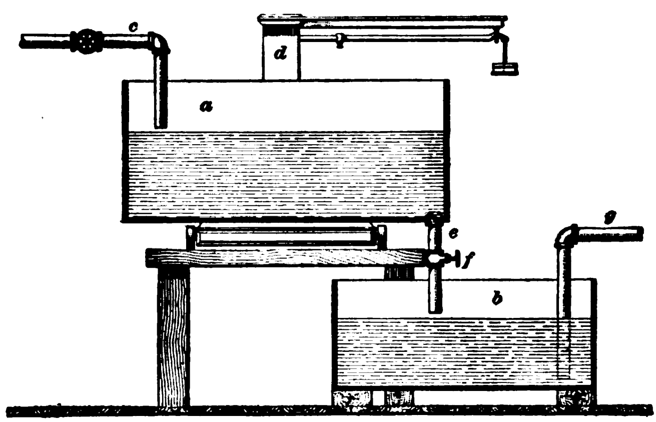


FIG. 4

the other. The supply of water is fed through the pipe *c* into the upper tank *a*, which rests on a platform scale *d*. After balancing the tank, the scale may be set to weigh 500 or 600 pounds of water, the water being run in until the beam rises, and then shut off. The upper tank is provided with a pipe *e* and valve *f*, by means of which the water may be discharged into the lower tank, from which it is fed to the boiler through the pipe *g*.

42. The attendant who keeps the record of coal supply or water supply should become accustomed to making the tally on his blank just before or after some regular operation. For example, the person who weighs the feedwater should record each tankful, say immediately after closing the valve in the supply pipe, or perhaps after emptying the upper tank into the lower. If this precaution is not observed, the attendant is liable to become uncertain as to whether he has recorded the previous data, and a mistake is almost certain to result.

HORSEPOWER AND EFFICIENCY TESTS

METHOD OF PROCEDURE

43. Standard Boiler Horsepower.—When making a horsepower or efficiency test, a more elaborate method of procedure is required than for a comparative fuel-value test. The reason for this is that different boilers generate steam at different pressures, different feedwater temperatures, and different degrees of dryness; hence, to compare the performances of boilers so as to determine their comparative efficiencies, it is necessary to reduce the actual evaporation to an equivalent evaporation from and at 212° F. per pound of combustible.

In order to have an accurate standard of boiler power, a committee of the American Society of Mechanical Engineers has recommended as a commercial horsepower *an evaporation of 30 pounds of water per hour from a feedwater temperature of 100° F. into steam at 70 pounds gauge pressure*, which is equivalent to $34\frac{1}{2}$ units of evaporation; that is, to $34\frac{1}{2}$ pounds of water evaporated from a feedwater temperature of 212° F. into steam at the same temperature.

Since 965.8 heat units are required to evaporate a pound of water from and at 212°, a horsepower is equal to $965.8 \times 34\frac{1}{2} = 33{,}320$ British thermal units per hour.

44. Equivalent Evaporation.—The **equivalent evaporation** is readily determined by means of the formula

$$W_1 = \frac{W(H - t + 32)}{965.8}$$

in which W = actual evaporation, in pounds of water per hour;

H = total heat of steam above 32° F. at observed pressure of evaporation;

t = observed feedwater temperature;

W_1 = equivalent evaporation, in pounds of water per hour, from and at 212° F.

EXAMPLE.—A boiler generates 2,200 pounds of dry steam per hour at a pressure of 120 pounds, gauge. The temperature of the feedwater being 70° F.: (*a*) what is the equivalent evaporation? (*b*) what is the horsepower of the boiler?

SOLUTION.—(*a*) According to the Steam Table, the total heat H corresponding to a gauge pressure of 120 lb. is 1,188.6 B. T. U. Applying the formula,

$$W_1 = \frac{2{,}200 \times (1{,}188.6 - 70 + 32)}{965.8} = 2{,}621 \text{ pounds. Ans.}$$

(*b*) The horsepower, which is obtained by dividing the total equivalent evaporation by 34.5, the equivalent of 1 H. P. is,

$$2{,}621 \div 34.5 = 76 \text{ H. P., nearly. Ans.}$$

45. Factor of Evaporation.—The quantity $\frac{H - t + 32}{965.8}$ that changes the actual evaporation of 1 pound of water to equivalent evaporation from and at 212° F. is called the **factor of evaporation.** To facilitate calculation of the equivalent evaporation, the table of factors of evaporation is inserted. The equivalent evaporation is found by multiplying the actual evaporation by the factor of evaporation taken from the table.

EXAMPLE 1.—A boiler is required to furnish 1,800 pounds of steam per hour at a gauge pressure of 80 pounds; if the temperature of the feedwater is 48° F., what will be the rated horsepower of the boiler?

SOLUTION.—From Table I, the factor of evaporation for 80 lb. pressure and a feedwater temperature of 40° is 1.214, and for the same pressure and a feedwater temperature of 50°, 1.203; the difference is 1.214 − 1.203 = .011. The difference of temperature is 50° − 40° = 10°.

TABLE I

FACTORS OF EVAPORATION

Temperature of Feedwater Degrees F.	Gauge Pressures															
	25	30	35	40	45	50	60	70	80	90	100	120	140	160	180	200
	Factors of Evaporation															
32	1.204	1.206	1.209	1.211	1.212	1.214	1.217	1.219	1.222	1.224	1.227	1.231	1.234	1.237	1.239	1.241
40	1.196	1.198	1.201	1.203	1.204	1.206	1.209	1.211	1.214	1.216	1.219	1.223	1.226	1.229	1.231	1.233
50	1.185	1.187	1.190	1.192	1.193	1.195	1.198	1.200	1.203	1.205	1.208	1.212	1.215	1.218	1.220	1.222
60	1.175	1.177	1.180	1.182	1.183	1.185	1.188	1.190	1.193	1.195	1.198	1.202	1.205	1.208	1.210	1.212
70	1.165	1.167	1.170	1.172	1.173	1.175	1.178	1.180	1.183	1.185	1.188	1.192	1.195	1.198	1.200	1.202
80	1.154	1.156	1.159	1.161	1.162	1.164	1.167	1.169	1.172	1.174	1.177	1.181	1.184	1.187	1.189	1.191
90	1.144	1.146	1.149	1.151	1.152	1.154	1.157	1.159	1.162	1.164	1.167	1.171	1.174	1.177	1.179	1.181
100	1.134	1.136	1.139	1.141	1.142	1.144	1.147	1.149	1.152	1.154	1.157	1.161	1.164	1.167	1.169	1.171
110	1.123	1.125	1.128	1.130	1.131	1.133	1.136	1.138	1.141	1.143	1.146	1.150	1.153	1.156	1.158	1.160
120	1.113	1.115	1.118	1.120	1.121	1.123	1.126	1.128	1.131	1.133	1.136	1.140	1.143	1.146	1.148	1.150
130	1.102	1.104	1.107	1.109	1.110	1.112	1.115	1.117	1.120	1.122	1.125	1.129	1.132	1.135	1.137	1.139
140	1.092	1.094	1.097	1.099	1.100	1.102	1.105	1.107	1.110	1.112	1.115	1.119	1.122	1.125	1.127	1.129
150	1.082	1.084	1.087	1.089	1.090	1.092	1.095	1.097	1.100	1.102	1.105	1.109	1.112	1.115	1.117	1.119
160	1.071	1.073	1.076	1.078	1.079	1.081	1.084	1.086	1.089	1.091	1.094	1.098	1.101	1.104	1.106	1.108
170	1.061	1.063	1.066	1.068	1.069	1.071	1.074	1.076	1.079	1.081	1.084	1.088	1.091	1.094	1.096	1.098
180	1.050	1.052	1.055	1.057	1.058	1.060	1.063	1.065	1.068	1.070	1.073	1.077	1.080	1.083	1.085	1.087
190	1.040	1.042	1.045	1.047	1.048	1.050	1.053	1.055	1.058	1.060	1.063	1.067	1.070	1.073	1.075	1.077
200	1.030	1.032	1.035	1.037	1.038	1.040	1.043	1.045	1.048	1.050	1.053	1.057	1.060	1.063	1.065	1.067
210	1.020	1.022	1.025	1.027	1.028	1.030	1.033	1.035	1.038	1.040	1.043	1.047	1.050	1.053	1.055	1.057

Difference between the lower temperature and the required temperature is $48° - 40° = 8°$. Then, $10° : 8° = .011 : x$, or $x = .009$; $1.214 - .009 = 1.205$. $1,800 \times 1.205 = 2,169$ lb., and

$$2,169 \div 34.5 = 63 \text{ H. P., nearly. Ans.}$$

EXAMPLE 2.—What is the factor of evaporation when the feedwater temperature is 122° F. and the gauge pressure 72 pounds?

SOLUTION.—In Table I, under the column headed 70 and opposite 120 in the left-hand column is found 1.128; in column headed 80 and opposite 120 is found 1.131; difference is .003. In the same vertical columns and opposite 130 are found 1.117 and 1.120; difference is .003, same as above. Hence, for an increase of 10 lb. in gauge reading, there is an increase of .003 in the factor of evaporation, or an increase of .0003 for 1 lb. and of $.0003 \times 2 = .0006$ for 2 lb. Therefore, for a feedwater temperature of 120° and 72 lb. pressure, the factor of evaporation is $1.128 + .0006 = 1.1286$. The difference between the numbers opposite 120 and 130 in the two columns headed 70 and 80, respectively, is $1.128 - 1.117 = .011$, and $1.131 - 1.120 = .011$, showing that, for an increase of temperature in the feedwater of 10°, there is a decrease in the factor of .011, and for 1° a decrease of .0011, or for 2° of .0022. Hence, the value of the factor for a temperature of 122° and a pressure (gauge) of 72 lb. is

$$1.1286 - .0022 = 1.126. \text{ Ans.}$$

46. Boiler Efficiency.—The **efficiency** of a boiler may be defined as the ratio of the heat utilized in evaporating water to the total heat supplied by the fuel. The efficiency thus calculated is really the combined efficiency of the furnace and boiler, as it is not easily possible to determine separately the efficiency of each.

The amount of heat supplied is determined by first accurately weighing the fuel used during the test and deducting all the ash and unconsumed portions. This weight, in pounds, is multiplied by the total heat of combustion of 1 pound of the combustible, as determined by an analysis, the product being the total number of heat units supplied during the test under the assumption that combustion was perfect. The heat usefully expended in evaporating water is obtained by first weighing the feedwater and correcting this weight according to the quality of the steam; the corrected weight is then multiplied by the number of heat units required to change water at the temperature of the feed into steam at the observed pressure.

The percentage of efficiency of a boiler may be found by the following formula,

$$E = \frac{100A}{B}$$

in which E = efficiency of boiler;
A = heat utilized in evaporating water;
B = total heat supplied by fuel.

EXAMPLE.—A boiler trial shows a useful expenditure of 186,429,030 British thermal units and a total supply of 270,187,000 British thermal units; what is the efficiency of the boiler?

SOLUTION.—Applying the formula,

$$E = \frac{100 \times 186,429,030}{270,187,000} = 69 \text{ per cent. Ans.}$$

DETERMINATION OF QUALITY OF STEAM

CALORIMETRIC TESTS

47. Quality of Steam.—In making a boiler trial, it is important to determine as closely as possible how much moisture, if any, the steam contains. Many boilers, especially when generating steam rapidly, furnish wet steam. The water is carried along with the steam in the form of spray or even in drops. Of course, this water is not evaporated, and if allowance is not made for it, the apparent efficiency of the boiler will be greater than its actual efficiency.

By the expression **quality of steam** is meant the percentage of the water fed into the boiler that is evaporated into pure dry steam. For example, suppose that for every 100 pounds of water fed to the boiler 98 pounds is changed to dry steam and 2 pounds is carried over in the form of water. Then, the quality of steam is 98 per cent. and the percentage of moisture is 2 per cent.

48. Barrel Calorimeter.—The quality of the steam must be determined by the use of a calorimeter. The **barrel calorimeter** is quite commonly used for this purpose, though great care must be exercised in operating it if trustworthy results are to be obtained. The method of procedure is as follows: A barrel or tank *a*, Fig. 5, holding 400 or 500 pounds of water, is placed on a platform scale *b*, and filled with water and weighed. The temperature of the water is registered by a thermometer inserted in the side of the barrel. Steam from the boiler is led through a pipe or hose *c* into the barrel until the temperature of the water reaches 130° or 140° F. The steam is then turned off and the barrel and contents are again weighed. The average steam pressure throughout the observation must be noted. It is also well to have the tube bent as shown in the figure.

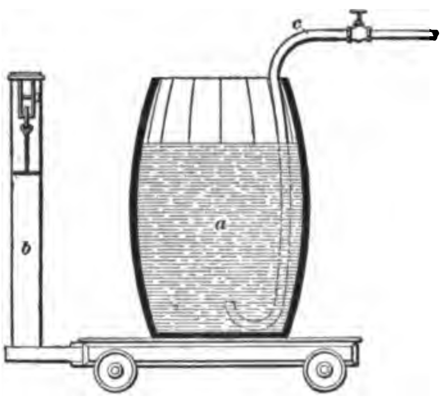


FIG. 5

Knowing now the weight of the cold water in the barrel, the weight of steam run in, the initial and final temperatures of the water, and the steam pressure, the quality of the steam may readily be found as follows:

Let W = original weight of water in barrel;
w = weight of steam and water that flow into the cold water;
l = latent heat of steam at observed pressure;
t = temperature of steam at observed pressure;
t_1 = initial temperature of water;
t_2 = final temperature of water;
x = portion of w that is dry steam;
Q = quality of steam, that is, percentage of mixture evaporated, which is pure dry steam.
S = superheat in degrees F.

The x pounds of dry steam give up $x\,l$ heat units on changing from steam to water; the combined steam and water w becomes lowered in temperature from t to t_2 on becoming mixed with the cold water in the barrel. Hence, the total number of heat units given up to the water in the barrel is

$$x\,l + w\,(t - t_2)$$

and this heat raises the temperature of the cold water from t_1 to t_2. Hence,

$$x\,l + w\,(t - t_2) = W(t_2 - t_1)$$

or
$$x = \frac{W(t_2 - t_1) - w\,(t - t_2)}{l}$$

and
$$Q = \frac{x}{w} = \frac{1}{l}\left[\frac{W}{w}(t_2 - t_1) - (t - t_2)\right] \qquad (1)$$

When Q is found to be greater than 100 per cent., it is evident that some of the heat that, in the formula, is assumed to have been used only in evaporating the water has actually been expended in superheating the steam. The amount of heat used in superheating is equal to the per cent. of superheat $(Q - 1)$ multiplied by the latent heat of steam at the observed pressure, or $(Q - 1)l$ British thermal units. The number of degrees of superheat is found by dividing this quantity of heat by .48, the specific heat of superheated steam. Thus, the amount of superheat is

$$S = \frac{(Q - 1)l}{.48} \text{ degrees F.} \qquad (2)$$

Example.—In a calorimetric test, the weight of cold water was 420 pounds, and of steam condensed was 36 pounds. The initial temperature of cold water was 40° F., the final temperature was 130° F., and the steam pressure was 60 pounds. Find the quality of the steam.

Solution.—By formula 1,

$$Q = \frac{1}{l}\left[\frac{W}{w}(t_2 - t_1) - (t - t_2)\right]$$

$$= \frac{1}{899}\left[\frac{420}{36}(130 - 40) - (307.1 - 130)\right] = 97.1 \text{ per cent., nearly}$$

In this formula, the values of l and t are obtained from the Steam Table. The boiler therefore generates a mixture that is composed of 97.1 per cent. dry steam and 2.9 per cent. water. Ans.

The one point in favor of the barrel calorimeter is its availability. A barrel, a platform scale, a length of hose, and a fairly good thermometer can be procured without trouble or great expense.

49. For refined measurements of the quality of steam, more accurate instruments must be used. The greatest care must be exercised in reading the thermometer (which, for this purpose, should read to tenths of a degree) and in weighing the water. The scales should be as finely graduated as possible, and so should the gauge that records the steam pressure. If possible, a barometer should be used to obtain the exact atmospheric pressure, which should be added to the gauge reading in order to obtain the absolute pressure for use in determining l and t when the Steam Table is employed. Slight errors in weighing and reading the thermometer and gauges will make a considerable difference in the value of Q. For example, suppose that the observed readings were $W = 200.5$ pounds, $w = 10.1$ pounds, steam pressure (gauge) $p = 78$ pounds, $t_1 = 44.5°$, and $t_2 = 100.5°$. Suppose, further, that the true readings should have been $W = 200$ pounds, $w = 10$ pounds, $p = 80$ pounds, $t_1 = 45°$, and $t_2 = 100°$. Substituting in formula **1**, Art. **48**,

$$Q = \frac{1}{l}\left[\frac{W}{w}(t_2 - t_1) - (t - t_2)\right],$$

the following results are obtained:

		Per Cent. Moisture	Error
For the true readings, the value of	$Q =$ .9881	1.19	0.00
For all readings true except $W = 200.5$,	$Q =$ .9912	.88	.31
For all readings true except $w = 10.1$,	$Q =$ .9758	2.41	1.22
For all readings true except $p = 78.0$,	$Q =$ .9886	1.15	.04
For all readings true except $t_1 = 44.5$,	$Q =$ .9994	.06	1.13
For all readings true except $t_2 = 100.5$,	$Q =$ .9999	.01	1.18
For all readings incorrect,	$Q =$ 1.0023	− .23	1.42

The last case indicates that the steam has been superheated 4.25°, since, by formula **2**, Art. **48**,

$$S = \frac{(Q-1)l}{.48} = \frac{(1.0023 - 1) \times 888}{.48} = 4.25°$$

50. Separator Calorimeter.—While the barrel calorimeter is probably the most available form of calorimeter, some doubt attaches to the results obtained by its use, and it has been the aim of some engineers to design a form of instrument for measuring the quality of the steam that would combine simplicity with reliability and be more trustworthy than the barrel calorimeter.

In the so-called **separator calorimeter** shown in Fig. 6, the steam to be tested passes through the pipe *a* and head *b* into the mechanical separator *c*. The steam escapes through a series of fine holes and passes over the edge of the cup *d* into the outer space *e*. Thence it passes through the nozzle *f* and hose *g* to a condenser *h*, where it is condensed. The sudden change in direction of the flow of the steam in passing through the fine orifices in *c* causes the water that is suspended in the steam to be thrown into the cup *d*. The quantity thus caught in the cup is shown by the gauge glass *i*. The scale *j* is so graduated that each division indicates $\frac{1}{100}$ pound. The separator frees the steam from entrained water, so that only practically dry steam passes to the condenser.

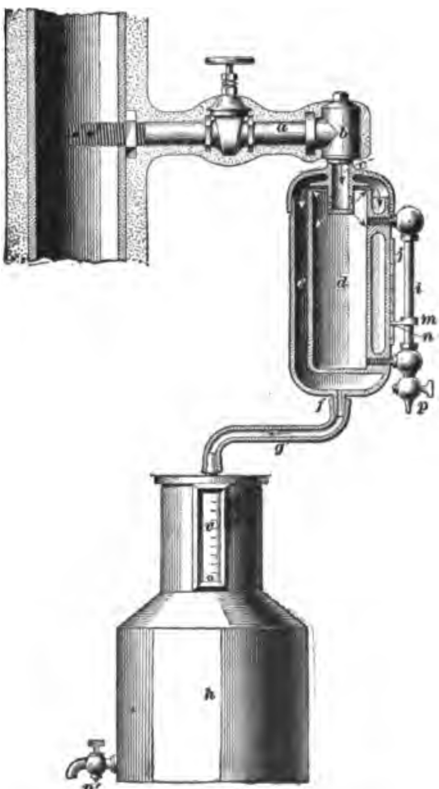


Fig. 6

The rate of flow of steam is limited by the size of the orifice in *f*, which is usually $\frac{1}{16}$ inch. Condensation with steam at 60 pounds gauge pressure will occur at the rate of about 1 pound in 3 minutes. The condenser should contain from 75 to 80 pounds of water, which should be as cold as can conveniently be obtained and should fill the condenser to or a little above the zero mark of the scale *v*. Each division of the scale *v* indicates $\frac{1}{10}$ pound. As the steam condenses, the water level will rise, and the difference between the successive readings of the scale indicates the weight of dry steam actually condensed. Readings should be taken simultaneously on the scales *v* and *j* at the beginning and end of each test. The weight of steam actually tested is the sum of the weights of the steam condensed in *h* and the water that is caught in *d*.

51. Before being used, the instrument must be warmed up slowly, to avoid breaking the gauge glass. The full pressure of steam should then be admitted and maintained until the test has been made. The valve and pipe connections should be well protected with good non-conducting material, but the body of the instrument and the condenser should be left uncovered.

With a separator calorimeter, the quality of the steam is found by the following formula:

$$Q = \frac{W}{W + w}$$

in which W = weight of condensed water;
w = weight of water in separator;
Q = quality of steam.

EXAMPLE.—The initial reading of the scale on the separator was .04 pound and the final reading .28 pound; the scale on the condenser indicated .6 pound at the beginning and 16.8 pounds at the ending of the test. What is the quality of the steam?

SOLUTION.—Water in separator = .28 − .04 = .24 lb. Water in condenser = 16.8 − .6 = 16.2 lb. Applying the above formula,

$$Q = \frac{16.2}{16.2 + .24} = .9854, \text{ or } 98.54 \text{ per cent. Ans.}$$

52. Throttling Calorimeter.—In the type of calorimeter shown in Fig. 7, steam enters through the pipe *a*, being throttled at the valve *b* as it enters the inner vessel *c*. A steam gauge *d* and thermometer *e* give the pressure and temperature of the steam. The outlet pipe *f* is open, so that the pressure in *c* does not rise very much above that of the atmosphere. As the steam passes the throttling valve *b*, the heat given up enters the water entrained in the steam and evaporates it, and whatever heat there may be in excess is utilized in superheating the steam.

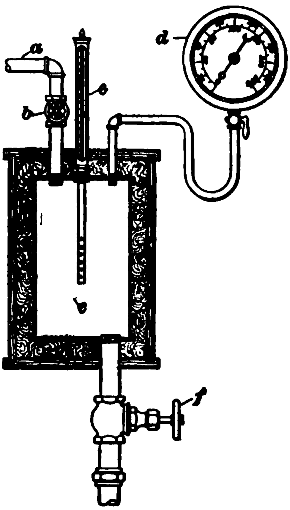


FIG. 7

Let Q = quality, or percentage of dry steam;

r = latent heat of steam at boiler pressure;

q = heat of liquid at boiler pressure;

t_1 = temperature of superheated steam in calorimeter;

t_2 = temperature of saturated steam at pressure in calorimeter;

H = total heat of saturated steam at pressure in calorimeter.

Then $$Qr + q = H + .48\,(t_1 - t_2);$$

therefore, $$Q = \frac{H + .48\,(t_1 - t_2) - q}{r}$$

EXAMPLE.—In a test with a throttling calorimeter the boiler pressure (gauge) was 125 pounds, the temperature of the superheated steam in the calorimeter 344° F., and the pressure in the calorimeter 106 pounds, gauge; what is the quality of the steam?

SOLUTION.—By the Steam Table, the latent heat r of steam for a boiler pressure of 125 lb. is 865.2 B. T. U., and the heat of the liquid q is 324.2 B. T. U. The temperature t_2 of saturated steam at 106 lb. gauge pressure is 341.48° and the total heat H is 1,186.1 B. T. U. Substituting in the formula of Art. **52**,

$$Q = \frac{1{,}186.1 + .48\,(344 - 341.48) - 324.2}{865.2} = .9976$$

Hence, the quality of the steam is 99.76%. Ans.

COAL ANALYSIS

METHODS OF ANALYZING COAL

53. There are two analyses of coal that may be made, the *ultimate analysis*, which gives the percentages of the various chemical elements of which the coal is composed, and the *proximate analysis*, which determines the percentages of moisture, volatile matter, fixed carbon, and ash, with a fair degree of accuracy. Each of these analyses furnishes information of considerable value to the engineer.

ULTIMATE ANALYSIS

54. The **ultimate analysis** shows that if a sample of coal is separated into its elements there will be found certain proportions of oxygen, hydrogen, carbon, etc. These proportions are generally expressed as percentages of the weight of the original sample, the weight of which is considered as a unit, or 100 per cent. From the ultimate analysis, the heating value of the coal may easily be estimated as follows:

Let h = heat of combustion of coal, per pound;
C = percentage of carbon;
O = percentage of oxygen;
H = percentage of hydrogen;
S = percentage of sulphur.

Then, $h = 14{,}600\,C + 62{,}000\left(H - \frac{O}{8}\right) + 4{,}000\,S$

EXAMPLE.—A coal contains 85 per cent. carbon, 4 per cent. oxygen, 6 per cent. hydrogen, 1 per cent. sulphur, and 4 per cent. ash; what is the heat of combustion per pound?

SOLUTION.—Applying the formula,

$$h = 14{,}600 \times .85 + 62{,}000\left(.06 - \frac{.04}{8}\right) + 4{,}000 \times .01 = 15{,}860 \text{ B. T. U.}$$
Ans.

To make an ultimate analysis of a fuel is a difficult and expensive operation that can be successfully performed only by a skilled chemist with the facilities of a well-appointed laboratory. Owing to its expense, it is seldom made except in such cases as important boiler trials, where it is desired to obtain the most accurate information possible regarding the properties of the fuel used.

PROXIMATE ANALYSIS

55. A **proximate analysis** of coal is nearly always required in connection with a boiler trial, and although the results obtained are, to a great extent, merely comparative, when the directions given are strictly followed, the results obtained are accurate enough to be of great service in determining the fuel value of coal. It is of the utmost importance that the directions given should be followed exactly, for slight variations in the method give large differences in the results. Since the results, so far as moisture, volatile combustible matter, and fixed carbon are concerned, are only comparative, they must be obtained under exactly the same conditions in every case if they are to be of any value.

56. Sampling.—In selecting a sample, about 5 pounds of the coal should be taken, exercising care, of course, to get a sample representing the whole quantity. Break this up and continue to quarter it, that is, divide the mass into successive quarters, until a sample weighing about 100 grams is left. Pulverize this and keep it in a tightly stoppered bottle until analyzed. The quartering and pulverizing should be carried out as rapidly as possible, to prevent either absorption or loss of moisture. The analysis should be made as soon as convenient after the sample is taken. Good results are obtained by the method of analysis given below, and it is probably more largely used than any other.

57. Moisture.—The following method of determining moisture is frequently used, although it differs somewhat from the method recommended by the American Society of Mechanical Engineers. Weigh 1 gram of the pulverized sample into a porcelain or platinum crucible. Place the crucible, uncovered, in an air bath having a temperature ranging from 104° to 107° C. (219.2° to 224.6° F.) and keep it at this temperature for exactly 1 hour. Place the crucible in a desiccator, or dryer, cover it, and allow it to cool. As soon as cool, weigh it, still covered, and find the loss in weight; this loss represents the moisture.

58. Volatile Combustible Matter.—Weigh 1 gram of the pulverized sample into a clean platinum crucible weighing 20 to 30 grams and having a tightly fitting cover. Such a cover, however, will not be tight enough to prevent the volatile matter escaping. Put on the cover and heat over a Bunsen burner for exactly 7 minutes. The burner should be adjusted so that it will give a good flame 20 centimeters (7.87 inches) high. The crucible should be supported, preferably, on a platinum triangle, with the bottom 7 centimeters (2.756 inches) above the top of the burner. The determination should be made in a place free from drafts. Cool the crucible in a desiccator, and weigh as soon as cool. From the total loss in weight subtract the amount of moisture found, and call the remainder volatile combustible matter. This determination should always be made on a fresh sample of coal, and not on the sample used for the determination of moisture.

59. Fixed Carbon and Ash.—After weighing the crucible for the determination of volatile combustible matter, draw the cover a little to one side, place the crucible in an inclined position on a triangle, place a good Bunsen burner under it, and heat until the carbon is completely burned off. This operation is likely to prove tedious, and may be hastened by letting the crucible cool from time to time, and by stirring the contents with a stout piece of platinum wire, taking care, of course, not to lose any of the material in the crucible while stirring it. Care must also be taken not to produce too strong a current of air in the crucible while heating it, as in this way particles may be carried out, and an incorrect value given to the coal by the apparent increase in fixed carbon and decrease in ash. When the residue in the crucible no longer shows any unburned carbon, heat it a few minutes longer, cool it in a desiccator or dryer, and then weigh. The difference between this weight and the last one is the weight of fixed carbon in the sample, and the substance remaining in the crucible is ash. The percentages of the different constituents are, of course, calculated in the usual manner, and as samples weighing 1 gram are taken the calculations are very simple. The sum of the percentages of fixed carbon and ash is, approximately, the percentage of coke that may be obtained from the coal.

60. The following are the weighings and steps of a proximate analysis:

MOISTURE

Sample	1.0000 gram
After heating	.9120 gram
Moisture	.0880 gram, or 8.8 per cent.

VOLATILE COMBUSTIBLE MATTER

Sample (less moisture) . .	.9120 gram
After heating	.7660 gram
Volatile combustible matter	.1460 gram, or 14.60 per cent.

FIXED CARBON

Residue from determination of volatile combustible matter	.7660 gram
After burning off carbon .	.1993 gram
Fixed carbon	.5667 gram, or 56.67 per cent.
Ash	.1993 gram, or 19.93 per cent.
	Total, 100.00 per cent.

61. Calculations From Analysis.—In making the proximate analysis, great care must be taken to weigh as exactly as possible, and also to avoid spilling any part of the sample. The composition of the hydrocarbons not being given by the proximate analysis, the heating value of the coal cannot be accurately computed from the results obtained by it; the average composition of the volatile combustibles, however, varies in most coals but little from the composition of marsh gas, CH_4; it will, therefore, be found that a very good approximate estimate of the heating value can be made by calculating from the percentages of fixed carbon and volatile hydrocarbons determined by the proximate analysis, under the assumption that all the volatile matter is composed of CH_4.

Under this assumption, the formula for calculating the approximate heating value from the percentages given by the proximate analysis is as follows:

$$h = 23{,}600\ V + 14{,}600\ C$$

where V is the percentage of hydrocarbons and the other letters have the same meaning as in the formula in Art. **16.**

EXAMPLE.—What is the heat of combustion of a coal having the following composition: moisture, .088; hydrocarbons, .146; fixed carbon, .5667; ash, .1993?

SOLUTION.—Applying the above formula,

$$h = 23{,}600 \times .146 + 14{,}600 \times .5667 = 11{,}719 \text{ B. T. U. Ans.}$$

ANALYSIS OF CHIMNEY GASES

62. Chimney- or flue-gas analyses have often served to indicate how the efficiency of boiler plants could be increased very materially without making complete engineering tests. Flue-gas analyses are of special value in determining heat losses due to the formation and escape of carbon monoxide as a result of imperfect combustion.

Samples of the gases to be analyzed are drawn from the furnace flue, or chimney through a **sampling tube** of platinum, porcelain, glass, or of metal cooled by water, care being taken to make an air-tight joint between the sampling tube and the opening into which it is inserted, using asbestos, plaster of Paris, putty, wet cotton waste, or other material for this purpose. Care must also be taken to stop up all crevices in the boiler setting or flue through which air might leak and thus affect the composition of the gases to be analyzed.

63. Fig. 8 shows a form of water-cooled, metal, gas-sampling tube consisting of an outer brass tube *a* surrounding another tube *b* through which is run the inner gas-collecting tube *c*. The tube *a* is about 3 feet long and $1\frac{1}{4}$ inches outside diameter, the inner tubes *b* and *c* having diameters of

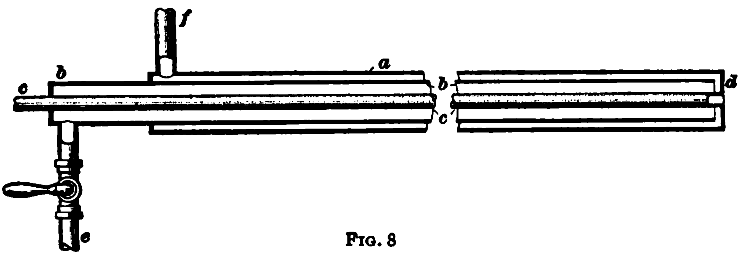


FIG. 8

$\frac{7}{8}$ and $\frac{1}{4}$ inch, respectively. The joint at the end *d* should be brazed, the others being soldered. Water enters the tube *b* through the pipe *e* and flows toward the end *d*, into the outer tube *a*, thence out through the pipe *f*.

64. The gases to be analyzed are usually drawn from the combustion chamber, chimney, or flue by means of some kind of water or steam-jet **aspirating pump,** one type of which is shown in Fig. 9. This pump somewhat resembles a boiler injector, its action in drawing the gases into the sampling apparatus depending on a flow of water from the pipe *a* through the constricted passage *b* into the waste tube *c*. As the water passes by the orifice *d* of the inspiration tube *e* the pressure in the latter is reduced and the air or gas therein flows toward the orifice *d*, through which the gas is sucked by the water, mingling with it and passing to waste through the tube *c*. The collecting bottle is placed between the aspirator or pump and the sampling tube. A light check-valve *f* serves to prevent the entrance of air

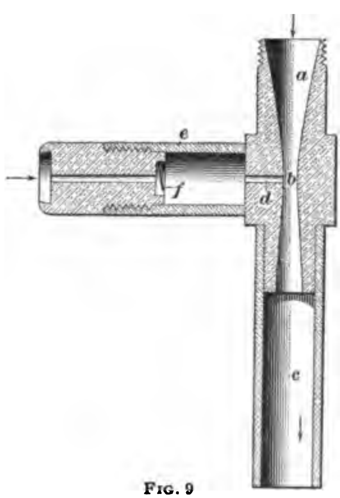


FIG. 9

when the gas is being drawn into the collecting bottle, or in case of stoppage of the water supply.

65. In order to secure samples that may safely be taken as approximately indicative of the general character of combustion, it is advisable to use a quart collecting bottle of the kind shown in Fig. 10, in which the collection of gas may go on continuously for ½ hour or longer. Through the stopper of the bottle are inserted a short piece of glass tubing *a* and a siphoning tube *b*. The gas sampling tube, Fig. 8, the aspirator, Fig. 9, and the collecting bottle are connected by means of the glass **T** *c*, Fig. 10, and short rubber-tubing connectors. When the bottle is connected as shown, the pinch cock *f* and screw cock on *d* are closed and the gas simply passes through the **T** *c* from the sampling tubes to the aspirator. Before collecting the sample, the air is expelled from the collecting bottle and the **T** by filling them with water. To fill the bottle with gas, the pinch cock and screw cock are opened slightly, and the water is slowly siphoned from the bottle through the tubes *b* and *d* while gas is being drawn into the bottle. The water flowing from the collecting bottle is thus replaced by the gases of combustion. The flow of water from the collecting bottle through the siphoning tube *b*, and consequently the rate at which the sample of gas is collected, may be regulated by means of the screw cock on the rubber tube *d*, so as to secure an average sample for any desired length of time. Preferably, the water used in the collecting bottle should first be saturated with the gas to be examined in order to reduce the possibilities of error in making analyses.

FIG. 10

66. Among the several types of appliances for analyzing the gases of combustion, that shown in Fig. 11 and known as the **Orsat** apparatus is one of the most popular. It generally consists of a burette *a*, which is simply a graduated glass tube for measuring the gas confined in it, connected to three or more glass **U**-shaped tubes or pipettes *h*, *i*, *j* containing chemical solutions for absorbing the different gaseous products of combustion, and a leveling bottle *m* for causing the gases to flow back and forth between the burette and pipettes. When the leveling bottle is raised, the water in it flows into the burette and causes the gas to pass into one of the pipettes. When the leveling bottle is lowered, the water in the burette runs back into the bottle, drawing the gas from the pipette back into the burette. To obviate errors due to temperature fluctuations that would otherwise affect the density of the gas, and hence the readings of the burette, the latter is enclosed, as shown, by a stoppered glass tube or jacket filled with water.

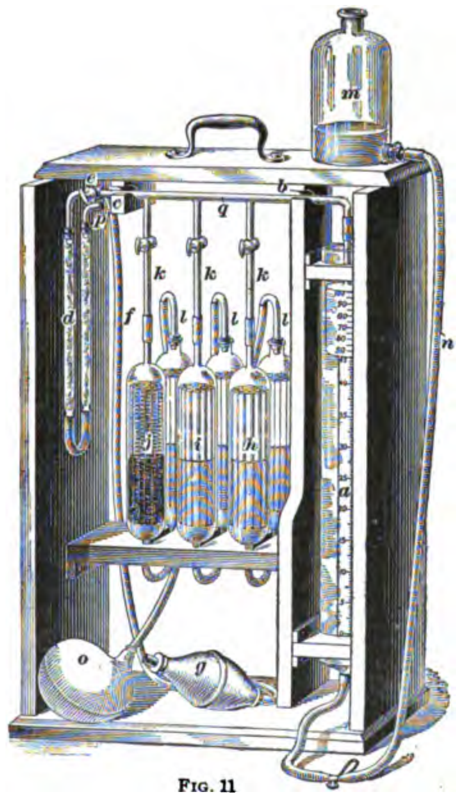


Fig. 11

The burette is connected to a thick tube *q*, fastened at *b* in a cut in the dividing panel and at *c* by means of a small brace attached to the cover of the case. The tube *q* bends down at its farther end and connects with a **U** tube *d* containing loose cotton in the arms, to catch dust, and water in the bend, the purpose of the water being to saturate the gas with moisture before measuring takes place. For increasing the absorbing surface of the pipettes, they are filled with glass tubes, as shown. Each glass tube in *j* contains a spiral of copper wire. The rear ends of the absorption pipettes are closed by rubber stoppers containing small glass tubes that are connected to a rubber ball *o* of about 200 cubic centimeters capacity, the purpose of the rubber being to effectively exclude air and prevent oxidation of the liquids.

In the boiler room, it has been found advisable to substitute rubber connections and pinch cocks for the glass stop-cocks in *k*, *k*, *k*, which, being fragile, are suitable for use in laboratory practice only. These stop-cocks are provided for the purpose of controlling the flow of gas into the pipettes. At *p*, where connection with the sampling tube or collecting bottle is made, it is advisable to provide a short rubber-tube connector that may be closed by a pinch cock. Piping connections between the Orsat and the sampling tube or the collecting bottle should be of tin, short pieces of rubber tubing being used to make joints.

In charging the pipettes, the front legs are opened to the atmosphere and the stoppers and tubes *l*, *l*, *l* are removed, and the chemical solutions are poured in until each pipette is about one-half full. In order that the solutions may be transferred from the rear legs of the pipettes to the front legs, where absorption of the various gases takes place, the three-way cock *e* is turned so as to establish communication with the atmosphere; then the burette is filled with water from the leveling bottle. The three-way cock *e* is now closed, the stop-cock in the tube to the pipette *h* is opened, the leveling bottle *m* is lowered, and as the water from the burette runs into the bottle the solution rises in the pipette *h*, and when it reaches the mark *k* the stop-cock is closed. The reagents in the other pipettes are raised to the marks *k*, *k*, *k* in the same way. The stoppers are then replaced and the tubes *l, l, l* are connected to the rubber bag *o*. No air can leak by the rubber-tube connectors between the tube *q* and the pipettes, because the marks *k, k, k* are above the connection, which is always moistened, and thus sealed by the solution in the pipette.

67. The apparatus now being ready for manipulation and the required sample of gas having been collected, the pinch cock *f* of the collecting bottle, Fig. 10, is closed, and the bottle is disconnected from the aspirator and sampling tube and connected with the Orsat at *p*. The burette having been filled with water, the air in the connecting tubes of the Orsat is then exhausted by means of a rubber suction bulb *g* attached by tubing to the three-way cock *e*; otherwise, after drawing the first sample from the collecting bottle into the burette, instead of pure gas there would be in the burette a mixture of air and gas that would have to be thrown away, and another sample would have to be drawn in order to obtain pure gas.

The air in the tubing connections between the collecting bottle and the pipettes and burette having been exhausted, the pinch cock *f*, Fig. 10, is opened and the three-way cock *e*, Fig. 11, is turned so as to establish communication between the burette and collecting bottle, from which gas is then drawn into the burette by lowering the leveling bottle *m* far enough to permit the water in the burette to run back into the bottle. Since the end of the tube *d*, Fig. 10, is immersed in the water siphoned from the collecting bottle, the gas withdrawn therefrom is replaced by the water that flows back into the collecting bottle under the influence of atmospheric pressure. After allowing the burette to drain for a couple of minutes, the cock *e* is closed and the leveling bottle is elevated until the water in the burette rises to the zero mark, whereupon the pinch cock on the tube *n*, Fig. 11, is closed and the three-way cock *e* is momentarily opened to relieve the gas in the burette of any pressure greater than that of the atmosphere, communication with which is then cut off by closing the cock *e*. The level of the water in the bottle *m* should then be brought to the same level as the water in the burette, namely, to zero, and the pinch cock *n* opened. No readings of the burette should at any time be taken until after the water in the burette and bottle is brought to the same level.

Having filled the burette with a good sample of gas, communication with the pipette *h*, containing a solution of caustic potash, KOH, for absorbing carbon dioxide, CO_2, is established by opening the cock, and the gas is forced from the burette into the pipette *h* by raising the leveling bottle *m*, so that the water therein will flow into the burette. By raising and lowering the leveling bottle, the gas is forced in and out of the pipette several times, the gas finally being drawn into the burette and the reading taken after the water levels in the burette and leveling bottle are brought to the same height. The first reading should be checked by a second, or even by a third or fourth reading, if necessary. The decrease of volume due to the absorption of carbon dioxide by the solution of caustic potash shows what percentage of the sample consists of carbon dioxide.

68. Having determined the amount of carbon dioxide present in a given volume of the gas, the gas in the burette is next forced into the pipette *i* that contains an alkaline solution of pyrogallate of potash for the absorption of oxygen. Any loss of volume that may appear after passing the gas in and out of the pipette *i* at least eight times will represent the volume of oxygen in the given volume of gas.

69. Finally, the gas is forced several times into and out of the pipette *j* containing an acid solution of cuprous chloride, $CuCl$, for the absorption of carbon monoxide, CO; when all the latter has been absorbed, that is, when the burette volume readings of successive trials agree exactly, no further diminution of volume being possible, the difference between the final reading and that obtained after passing the gas into the second pipette represents the volume of carbon monoxide present.

70. After passing through the absorption process described, the gas contains hydrogen, nitrogen, and compounds of hydrogen and carbon, or hydrocarbons, such as marsh gas, or methane, CH_4, the determination of the percentages of which is attended with too much difficulty to warrant an attempt at separation in the boiler room. Moreover, such determination is unnecessary, since the principal object of making flue-gas analyses is to ascertain whether or not combustion is complete in order that the engineer may know whether or not the fires are properly managed and what changes should be made to secure greater economy.

While using the apparatus, care should be taken to prevent drawing any of the solutions into the burette, and the latter should be permitted to drain thoroughly before readings are taken. A record should be kept to show the amount of gas absorbed by each of the solutions in order that they may be renewed before they become saturated. Temperature changes should be noted carefully; a change of 2° F. at the burette will affect its readings about .3 per cent.

71. The caustic-potash solution is prepared by dissolving 500 grams of potassium hydrate, KOH, in 1,000 grams of water.

The pyrogallate-of-potash solution is made by dissolving 120 grams of potassium hydrate, KOH, in 100 grams of water and pouring the solution over 5 grams of solid pyrogallic acid, which is thereby dissolved and absorbed.

The cuprous-chloride, $CuCl$, solution is prepared by covering the bottom of a 2-quart bottle with copper oxide (scale) to a depth of about $\frac{3}{8}$ inch and putting in the bottle a bundle of copper wires long enough to reach from top to bottom of the hydrochloric acid, HCL, that is next poured into the bottle. The bottle should be shaken occasionally, a fresh supply of hydrochloric acid, which should have a specific gravity of 1.10, being introduced whenever part of the solution is drawn off for use in the Orsat. Care should also be taken to renew the supply of copper oxide and wire whenever necessary, in order to insure a saturated solution. The before-named reagents possess absorptive powers approximately as follows: Caustic potash absorbs about 40 cubic centimeters of carbon dioxide, CO_2, per cubic centimeter; pyrogallate of potash about 22 cubic centimeters of oxygen, O, per cubic centimeter; cuprous chloride about 6 cubic centimeters of carbon monoxide, CO, per cubic centimeter.

STANDARD FORM OF BOILER TRIAL

72. The American Society of Mechanical Engineers in 1885 accepted the report of a committee that had formulated a set of rules for the conduct of boiler trials with the object of securing uniform results. The code of rules was revised in 1899 by another committee and the universal adoption of the revised rules recommended. The rules are given below practically in full.

RULES FOR CONDUCTING BOILER TRIALS

73. Code of 1899.—I. *Determine at the outset* the specific object of the proposed trial, whether it be to ascertain the capacity of the boiler, its efficiency as a steam generator, its efficiency and its defects under usual working conditions, the economy of some particular kind of fuel, or the effect of changes of design, proportion, or operation; and prepare for the trial accordingly.

II. *Examine the boiler*, both outside and inside; ascertain the dimension of grates, heating surfaces, and all important parts; and make a full record, describing the same, and illustrating special features by sketches. The area of heating surface is to be computed from the surfaces of shells, tubes, furnaces, and fireboxes in contact with the fire or hot gases. The outside diameter of water tubes and the inside diameter of fire-tubes are to be used in the computation. All surfaces below the mean water level which have water on one side and products of combustion on the other are to be considered as water-heating surface, and all surfaces above the mean water level which have steam on one side and products of combustion on the other are to be considered as superheating surface.

III. *Notice the general condition* of the boiler and its equipment, and record such facts in relation thereto as bear on the objects in view.

If the object of the trial is to ascertain the maximum economy or capacity of the boiler as a steam generator, the boiler and all its appurtenances should be put in first-class condition. Clean the heating surface inside and outside, remove clinkers from the grates and from the sides of the furnace. Remove all dust, soot, and ashes from the chambers, smoke connections, and flues. Close air leaks in the masonry and poorly fitted cleaning doors. See that the damper will open wide and close tight. Test for air leaks by firing a few shovels of smoky fuel and immediately closing the damper, observing the escape of smoke through the crevices, or by passing the flame of a candle over cracks in the brickwork.

IV. *Determine the character of the coal* to be used. For tests of the efficiency or capacity of the boiler for comparison with other boilers, the coal should, if possible, be of some kind which is commercially regarded as a standard. For New England and that portion of the country east of the Alleghany Mountains, good anthracite egg coal, containing not over 10 per cent. of ash, and semibituminous Clearfield (Pennsylvania), Cumberland (Maryland), and Pocahontas (Virginia) coals are thus regarded. West of the Alleghany Mountains, Pocahontas (Virginia) and New River (West Virginia) semibituminous, and Youghiogheny or Pittsburg bituminous coals are recognized as standards. There is no special grade of coal mined in the Western States which is widely recognized as of superior quality or considered as a standard coal for boiler testing. Big Muddy lump, an Illinois coal mined in Jackson County, Illinois, is suggested as being of sufficiently high grade to answer these requirements in districts where it is more conveniently obtainable than the other coals mentioned above.

For tests made to determine the performance of a boiler with a particular kind of coal, such as may be specified in a contract for the sale of a boiler, the coal used should not be higher in ash and in moisture than that specified, since increase in ash and moisture above a stated amount is apt to cause a falling off of both capacity and economy in greater proportion than the proportion of such increase.

V. *Establish the correctness of all apparatus* used in the test for weighing and measuring. These are:

1. Scales for weighing coal, ashes, and water.
2. Tanks or water meters for measuring water. Water meters, as a rule, should be used only as a check on other measurements. For accurate work, the water should be weighed or measured in a tank.
3. Thermometers and pyrometers for taking temperatures of air, steam, feedwater, waste gases, etc.
4. Pressure gauges, draft gauges, etc.

The kind and location of the various pieces of testing apparatus must be left to the judgment of the person conducting the test, always keeping in mind the main object; that is, to obtain authentic data.

VI. *See that the boiler is thoroughly heated* before the trial to its usual working temperature. If the boiler is new and of a form provided with a brick setting, it should be in regular use at least a week before the trial, so as to dry and heat the walls. If it has been laid off and become cold, it should be worked before the trial until the walls are well heated.

VII. The *boiler and connections* should be proved to be free from leaks before beginning a test, and all water connections, including blow-off and extra feedpipes, should be disconnected, stopped with blank flanges, or bled through special openings beyond the valves, except the particular pipe through which water is to be fed to the boiler during the trial. During the test, the blow-off and feedpipes should remain exposed to view. If an injector is used, it should receive steam directly through a felted pipe from the boiler being tested.

NOTE.—In feeding a boiler undergoing test with an injector taking steam from another boiler or from the main steam pipe from several boilers, the evaporative results may be modified by a difference in the quality of the steam from such source compared with that supplied by

the boiler being tested, and in some cases the connection to the injector may act as a drip for the main steam pipe. If it is known that the steam from the main steam pipe is of the same pressure and quality as that furnished by the boiler undergoing the test, the steam may be taken from such main steam pipe.

If the water is metered after it passes the injector, its temperature should be taken at the point where it leaves the injector. If the quantity is determined before it goes to the injector, the temperature should be determined on the suction side of the injector; and if no change of temperature occurs other than that due to the injector, the temperature thus determined is properly that of the feed-water. When the temperature changes between the injector and the boiler, as by the use of a heater or by radiation, the temperature at which the water enters and leaves the injector and that at which it enters the boiler should all be taken. In that case, the weight to be used is that of the water leaving the injector, computed from the heat units if not directly measured, and the temperature that of the water entering the boiler.

Let w = weight of water entering injector;
x = weight of steam entering injector;
h_1 = heat units per pound of water entering injector;
h_2 = heat units per pound of steam entering injector;
h_3 = heat units per pound of water leaving injector.

Then, $w + x$ = weight of water leaving injector;

$$x = w\frac{h_3 - h_1}{h_2 - h_3}$$

See that the steam main is so arranged that water of condensation cannot run back into the boiler.

VIII. *Duration of the Test.*—For tests made to ascertain either the maximum economy or the maximum capacity of the boiler, irrespective of the particular class of service for which it is regularly used, the duration should be at least 10 hours of continuous running.

If the rate of combustion exceeds 25 pounds of coal per square foot of grate surface per hour, it may be stopped when a total of 250 pounds of coal has been burned per square foot of grate.

In cases where the service requires continuous running for the whole 24 hours of the day, with shifts of firemen a number of times during that period, it is well to continue the test for at least 24 hours.

When it is desired to ascertain the performance under the working conditions of practical running, whether the boiler be regularly in use 24 hours a day or only a certain number of hours out of each 24, the fires being banked the balance of the time, the duration should not be less than 24 hours.

IX. *Starting and Stopping a Test.*—The conditions of the boiler and furnace in all respects should be, as nearly as possible, the same at the end as at the beginning of the test. The steam pressure should be the same; the water level the same; the fire on the grates should be the same in quantity and condition; and the walls, flues, etc. should be of the same temperature. Two methods of obtaining the desired equality of conditions of the fire may be used, viz.: "the standard method" and "the alternate method," the latter being employed where it is inconvenient to make use of the standard method.

X. *Standard Method of Starting and Stopping a Test.* Steam being raised to the working pressure, remove rapidly all the fire from the grate, close the damper, clean the ash-pit, and as quickly as possible start a new fire with weighed wood and coal, noting the time and the water level while the water is in a quiescent state, just before lighting the fire.

NOTE.—The gauge glass should not be blown out within an hour before the water level is taken at the beginning and end of a test, otherwise an error in the reading of the water level may be caused by a change in the temperature and density of the water in the pipe leading from the bottom of the glass into the boiler.

At the end of the test remove the whole fire, which has been burned low, clean the grates and ash-pit, and note the water level when the water is in a quiescent state, and record the time of hauling the fire. The water level should be as nearly as possible the same as at the beginning of the test. If it is not the same, a correction should be made by computation and not by operating the pump after the test is completed.

XI. *Alternate Method of Starting and Stopping a Test.* The boiler being thoroughly heated by a preliminary run, the fires are to be burned low and well cleaned. Note the amount of coal left on the grate as nearly as it can be estimated; note the pressure of steam and the water level. Note the time and record it as the starting time. Fresh coal which has been weighed should now be fired. The ash-pits should be thoroughly cleaned at once after starting. Before the end of the test, the fires should be burned low, just as before the start, and the fires cleaned in such a manner as to leave a bed of coal on the grates of the same depth and in the same condition as at the start. When this stage is reached, note the time and record it as the stopping time. The water level and steam pressure should previously be brought as nearly as possible to the same point as at the start. If the water level is not the same as at the start, a correction should be made by computation and not by operating the pump after the test is completed.

XII. *Uniformity of Conditions.*—In all trials made to ascertain maximum economy or capacity, the conditions should be maintained uniformly constant. Arrangements should be made to dispose of the steam so that the rate of evaporation may be kept the same from beginning to end. This may be accomplished in a single boiler by carrying the steam through a waste steam pipe, the discharge from which can be regulated as desired. In a battery of boilers, in which only one is tested, the draft may be regulated on the remaining boilers, leaving the test boiler to work under a constant rate of production.

Uniformity of conditions should prevail as to the pressure of steam, the height of water, the rate of evaporation, the thickness of fire, the times of firing and quantity of coal fired at one time, and as to the intervals between the times of cleaning the fires.

The method of firing to be carried on in such tests should be dictated by the expert or person in responsible charge of the test, and the method adopted should be adhered to by the fireman throughout the test.

XIII. *Keeping the Records.*—Take note of every event connected with the progress of the trial, however unimportant it may appear. Record the time of every occurrence and the time of taking every weight and every observation.

The coal should be weighed and delivered to the fireman in equal proportions, each sufficient for not more than 1 hour's run, and a fresh portion should not be delivered until the previous one has all been fired. The time required to consume each portion should be noted, the time being recorded at the instant of firing the last of each portion. It is desirable that at the same time the amount of water fed into the boiler should be accurately noted and recorded, including the height of the water in the boiler and the average pressure of steam and temperature of feed during the time. By thus recording the amount of water evaporated by successive portions of coal, the test may be divided into several periods, if desired, and the degree of uniformity of combustion, evaporation, and economy analyzed for each period. In addition to these records of the coal and the feedwater, half hourly observations should be made of the temperature of the feedwater, of the flue gases, of the external air in the boiler room, of the temperature of the furnace when a furnace pyrometer is used, also of the pressure of steam, and of the readings of the instruments for determining the moisture in the steam. A log should be kept on properly prepared blanks containing columns for record of the various observations.

When the "standard method" of starting and stopping the test is used, the hourly rate of combustion and of evaporation and the horsepower should be computed from the records taken during the time when the fires are in active condition. This time is somewhat less than the actual time which elapses between the beginning and end of the run. The loss of time due to kindling the fire at the beginning and burning it out at the end makes this course necessary.

XIV. *Quality of Steam.*—The percentage of moisture in the steam should be determined by the use of either a throttling or a separating steam calorimeter. The sampling nozzle should be placed in the vertical steam pipe rising from the boiler. It should be made of $\frac{1}{2}$-inch pipe and should extend across the diameter of the steam pipe to within $\frac{1}{2}$ inch of the opposite side, being closed at the end and perforated with not less than twenty $\frac{1}{8}$-inch holes equally distributed along and around its cylindrical surface, but none of these holes should be nearer than $\frac{1}{2}$ inch to the inner side of the steam pipe. The calorimeter and the pipe leading to it should be well covered with felting. Whenever the indications of the throttling or separating calorimeter show that the percentage of moisture is irregular or occasionally in excess of 3 per cent., the results should be checked by a steam separator placed in the steam pipe as close to the boiler as convenient, with a calorimeter in the steam pipe just beyond the outlet from the separator. The drip from the separator should be caught and weighed and the percentage of moisture computed therefrom added to that shown by the calorimeter.

Superheating should be determined by means of a thermometer placed in a mercury well inserted in the steam pipe. The degree of superheating should be taken as the difference between the reading of the thermometer for superheated steam and the readings of the same thermometer for saturated steam at the same pressure as determined by a special experiment and not by reference to Steam Tables.

XV. *Sampling the Coal and Determining Its Moisture.*—As each barrow load or fresh portion of coal is taken from the coal pile, a representative shovelful is selected from it and placed in a barrel or box in a cool place and kept until the end of the trial. The samples are then mixed and broken into pieces not exceeding 1 inch in diameter and reduced by the process of repeated quartering and crushing until a final sample weighing about 5 pounds is obtained and the size of the larger pieces is such that they will pass through a sieve with $\frac{1}{4}$-inch meshes. From this sample, two 1-quart, air-tight, glass preserving jars, or other air-tight vessels which will prevent the escape of moisture from the sample, are to be promptly filled, and these samples are to be kept for subsequent determinations of moisture and of heating value and for chemical analyses. During the process of quartering, when the sample has been reduced to about 100 pounds, a quarter to a half of it may be taken for an approximate determination of moisture. This may be made by placing it in a shallow iron pan, not over 3 inches deep, carefully weighing it, and setting the pan in the hottest place that can be found on the brickwork of the boiler setting or flues, keeping it there for at least 12 hours, and then weighing it. The determination of moisture thus made is believed to be approximately accurate for anthracite and semibituminous coals, and also for Pittsburg or Youghiogheny coal; but it cannot be relied on for coals mined west of Pittsburg or for other coals containing inherent moisture. For these latter coals, it is important that a more accurate method be adopted. The method recommended by the Committee for all accurate tests, whatever the character of the coal, is described as follows:

Take one of the samples contained in the glass jars and subject it to a thorough air-drying by spreading it in a thin layer and exposing it for several hours to the atmosphere of a warm room, weighing it before and after, thereby determining the quantity of surface moisture it contains. Then crush the whole of it by running it through an ordinary coffee mill adjusted so as to produce somewhat coarse grains (less than $\frac{1}{16}$ inch), thoroughly mix the crushed sample, select from it a portion of from 10 to 50 grams, weigh it in a balance which will easily show a variation as small as 1 part in 1,000, and dry it in an air or sand bath at a temperature between 240° and 280° F. for 1 hour. Weigh it and record the loss, then heat and weigh it again repeatedly at intervals for an hour or less, until the minimum weight has been reached and the weight begins to increase by oxidation of a portion of the coal. The difference between the original and the minimum weight is taken as the moisture in the air-dried coal. This moisture test should preferably be made on duplicate samples, and the results should agree within .3 to .4 of 1 per cent., the mean of the two determinations being taken as the correct result. The sum of the percentage of moisture thus found and the percentage of surface moisture previously determined is the total moisture.

XVI. *Treatment of Ashes and Refuse.*—The ashes and refuse are to be weighed in a dry state. If it is found desirable to show the principal characteristics of the ash, a sample should be subjected to a proximate analysis and the actual amount of incombustible material determined. For elaborate trials, a complete analysis of the ash and refuse should be made.

XVII. *Calorific Tests and Analysis of Coal.*—The quality of the fuel should be determined either by heat test or by analysis, or by both.

The rational method of determining the total heat of combustion is to burn the sample of the coal in an atmosphere of oxygen gas, the coal to be sampled as directed in Article XV of this code.

The chemical analysis of the coal should be made only by an expert chemist. The total heat of combustion computed from the results of the ultimate analysis may be obtained by the use of Dulong's formula (with constants modified by recent determinations); viz., $14{,}600\,C + 62{,}000\left(H - \frac{O}{8}\right) + 4{,}000\,S$, in which C, H, O, and S refer to the proportions of carbon, hydrogen, oxygen, and sulphur, respectively, as determined by the ultimate analysis.

It is desirable that a proximate analysis should be made, thereby determining the relative proportions of volatile matter and fixed carbon. These proportions furnish an indication of the leading characteristics of the fuel and serve to fix the class to which it belongs. As an additional indication of the characteristics of the fuel, the specific gravity should be determined.

XVIII. *Analysis of Flue Gases.*—The analysis of the flue gases is an especially valuable method of determining the relative value of different methods of firing or of different kinds of furnaces. In making these analyses, great care should be taken to procure average samples, since the composition is apt to vary at different points of the flue. The composition is also apt to vary from minute to minute, and for this reason the drawings of gas should last a considerable period of time. Where complete determinations are desired, the analyses should be intrusted to an expert chemist.

For the continuous indication of the amount of carbonic acid (carbon dioxide) present in the flue gases, an instrument may be employed which shows the weight of the sample of gas passing through it.

XIX. *Smoke Observations.*—It is desirable to have a uniform system of determining and recording the quantity of smoke produced where bituminous coal is used. The system commonly employed is to express the degree of smokiness by means of percentages dependent on the judgment of the observer. The Committee does not place much value on the percentage method, because it depends so largely on the personal element, but if this method is used it is desirable that so far as possible a definition be given in explicit terms as to the basis and method employed in arriving at the percentage. The actual measurement of a sample of soot and smoke by some form of meter is to be preferred.

XX. *Miscellaneous.*—In tests for purposes of scientific research, in which the determination of all the variables entering into the test is desired, certain observations should be made which are, in general, unnecessary for ordinary tests. These are the measurements of the air supply, the determination of its contained moisture, the determination of the amount of heat lost by radiation, of the amount of infiltration of air through the setting, and (by condensation of all the steam made by the boiler) of the total heat imparted to the water.

As these determinations are rarely undertaken, it is not deemed advisable to give directions for making them.

XXI. *Calculations of Efficiency.*—Two methods of defining and calculating the efficiency of a boiler are recommended.

They are:

1. Efficiency of the boiler

$$= \frac{\text{heat absorbed per pound combustible}}{\text{calorific value of 1 pound combustible}}$$

2. Efficiency of the boiler and grate

$$= \frac{\text{heat absorbed per pound coal}}{\text{calorific value of 1 pound coal}}$$

The first of these is sometimes called the efficiency based on combustible and the second the efficiency based on coal. The first is recommended as a standard of comparison for all tests, and this is the one which is understood to be referred to when the word "efficiency" alone is used without qualification. The second, however, should be included in a report of a test, together with the first, whenever the object of the test is to determine the efficiency of the boiler and furnace together with the grate (or mechanical stoker), or to compare different furnaces, grates, fuels, or methods of firing. The heat absorbed per pound of combustible (or per pound of coal) is to be calculated by multiplying the equivalent evaporation from and at 212° per pound combustible (or coal) by 965.7.

XXII. *The Heat Balance.*—An approximate "heat balance," or statement of the distribution of the heating value of the coal among the several items of heat utilized and heat lost, may be included in the report of a test when analyses of the fuel and of the chimney gases have been made. It should be reported in the form shown on the following page.

XXIII. *Report of the Trial.*—The data and results should be reported in the manner given in either one of the two following tables, omitting lines where the tests have not been made as elaborately as provided for in such tables. Additional lines may be added for data relating to the specific object of the test. The extra lines should be classified under the headings provided in the tables and numbered as per preceding line, with sub. letters *a*, *b*, etc. The Short Form of Report, Table No. 2, is recommended for commercial tests and as a convenient form of abridging the longer form for publication when saving of space is

HEAT BALANCE, OR DISTRIBUTION OF THE HEATING VALUE OF THE COMBUSTIBLE

Total Heat Value of 1 Pound of Combustible________B. T. U.

	B. T. U.	Per Cent.
1. Heat absorbed by the boiler = evaporation from and at 212° per pound of combustible × 965.7		
2. Loss due to moisture in coal = per cent. of moisture referred to combustible ÷ 100 × $[(212 - t) + 966 + .48\ (T - 212)]$. ($t$ = temperature of air in the boiler room, T = that of the flue gases)		
3. Loss due to moisture formed by the burning of hydrogen = per cent. of hydrogen to combustible ÷ 100 × 9 × $[(212 - t) + 966 + .48\ (T - 212)]$		
4. Loss due to heat carried away in the dry chimney gases = weight of gas per pound of combustible × .24 × $(T - t)$		
5. Loss due to incomplete combustion of carbon = $\dfrac{CO}{CO_2 + CO} \times \dfrac{\text{per cent. } C \text{ in combustible}}{100} \times 10{,}150$		
6. Loss due to unconsumed hydrogen and hydrocarbons, to heating the moisture in the air, to radiation, and unaccounted for. (Some of these losses may be separately itemized if data are obtained from which they may be calculated.)		
Totals		100.00

desirable. For elaborate trials, it is recommended that the full log of the trial be shown graphically by means of a chart.

In Table No. 1, the items printed in Italics correspond to the items in Table No. 2.

TABLE NO. 1

DATA AND RESULTS OF EVAPORATIVE TEST

ARRANGED IN ACCORDANCE WITH THE COMPLETE FORM ADVISED BY THE BOILER-TEST COMMITTEE OF THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS. CODE OF 1899.

Made by__________on__________boiler, at__________
to determine____________________

Principal conditions governing the trial____________________

*Kind of fuel*____________________
*Kind of furnace*____________________
State of the weather____________________
*Method of starting and stopping the test ("standard" or "alternate" Arts. X and XI, Code)*__________

1. *Date of trial*____________________
2. *Duration of trial*____________________hours

DIMENSIONS AND PROPORTIONS

A complete description of the boiler and drawings of the same, if of unusual type, should be given on an annexed sheet.

3. *Grate surface*__________*width*__________
   *length*__________*area*__________sq. ft.
4. Height of furnace . . . . in.
5. Approximate width of air spaces in grate . . . in.
6. Proportion of air space to whole grate surface . . . . per cent.
7. *Water-heating surface* . . . . sq. ft.
8. *Superheating surface* . . . . sq. ft.

9.	Ratio of water-heating surface to grate surface	to 1
10.	Ratio of minimum draft area to grate surface	1 to

Average Pressures

11.	*Steam pressure by gauge*	lb. per sq. in.
12.	*Force of draft between damper and boiler*	in. of water
13.	Force of draft in furnace	in. of water
14.	Force of draft or blast in ash-pit . . .	in. of water

Average Temperatures

15.	Of external air	deg.
16.	Of fireroom	deg.
17.	Of steam	deg.
18.	Of feedwater entering heater	deg.
19.	Of feedwater entering economizer	deg.
20.	*Of feedwater entering boiler*	deg.
21.	*Of escaping gases from boiler*	deg.
22.	Of escaping gases from economizer	deg.

Fuel

23.	Size and condition	
24.	Weight of wood used in lighting fire	lb.
25.	*Weight of coal as fired**	lb.
26.	*Percentage of moisture in coal*	per cent.
27.	*Total weight of dry coal consumed*	lb.
28.	*Total ash and refuse*	lb.
29.	Quality of ash and refuse	
30.	Total combustible consumed	lb.
31.	*Percentage of ash and refuse in dry coal*	per cent.

Proximate Analysis of Coal

		Of Coal	Of Combustible
32.	Fixed carbon	per cent.	per cent.
33.	Volatile matter	per cent.	per cent.
34.	Moisture	per cent.	
35.	Ash	per cent.	
		100 per cent.	100 per cent.
36.	Sulphur, separately determined	per cent.	per cent.

*Including Item 24 multiplied by .4.

Ultimate Analysis of Dry Coal
(*Art. XVII, Code*)

		Of Coal	Of Combustible
37.	Carbon, C	per cent.	per cent.
38.	Hydrogen, H	per cent.	per cent.
39.	Oxygen, O	per cent.	per cent.
40.	Nitrogen, N	per cent.	per cent.
41.	Sulphur, S	per cent.	per cent.
42.	Ash	per cent.	
		100 per cent.	100 per cent.
43.	Moisture in sample of coal as received	per cent.	per cent.

Analysis of Ash and Refuse

44.	Carbon	per cent.
45.	Earthy matter	per cent.

Fuel Per Hour

46.	*Dry coal consumed per hour*	lb.
47.	Combustible consumed per hour	lb.
48.	*Dry coal per square foot of grate surface per hour* .	lb.
49.	Combustible per square foot of water-heating surface per hour	lb.

Calorific Value of Fuel
(*Art. XVII, Code*)

50.	*Calorific value by oxygen calorimeter, per pound of dry coal*	B. T. U.
51.	*Calorific value by oxygen calorimeter, per pound of combustible*	B. T. U.
52.	Calorific value by analysis, per pound of dry coal	B. T. U.
53.	Calorific value by analysis, per pound of combustible	B. T. U.

Quality of Steam

54.	*Percentage of moisture in steam*	per cent.
55.	*Number of degrees of superheating*	deg.
56.	Quality of steam (dry steam = unity) . .	

Water

57. *Total weight of water fed to boiler* . . . . lb.
58. Equivalent water fed to boiler from and at 212° lb.
59. *Water actually evaporated, corrected for quality of steam* . . . . lb.
60. Factor of evaporation* . . . . lb.
61. Equivalent water evaporated into dry steam from and at 212° (Item 59 times Item 60) . . lb.

Water Per Hour

62. *Water evaporated per hour, corrected for quality of steam* . . . . lb.
63. *Equivalent evaporation per hour from and at 212°* lb.
64. *Equivalent evaporation per hour from and at 212° per square foot of water-heating surface* . . . . lb.

Horsepower

65. *Horsepower developed.* (34½ *pounds of water evaporated per hour into dry steam from and at 212° equals 1 horsepower)* . . . . H. P.
66. *Builders' rated horsepower* . . . . H. P.
67. *Percentage of builders' rated horsepower developed* . . . . per cent.

Economic Results

68. *Water apparently evaporated under actual conditions per pound of coal as fired.* (*Item 57 divided by Item 25*) . . . . lb.
69. *Equivalent evaporation from and at 212° per pound of coal as fired.* (*Item 61 divided by Item 25*). . . . . lb.
70. *Equivalent evaporation from and at 212° per pound of dry coal.* (*Item 61 divided by Item 27*) lb.

---

*The factor of evaporation is to be computed here from the formula $f = \frac{H - t + 32}{965.7}$, where f is factor of evaporation; H, total heat of steam; and t, the feedwater temperature. The factor of evaporation thus calculated will differ slightly from that calculated by the formula in Art. 45, on account of the Committee having taken 965.7 B. T. U. as the latent heat of steam at 212° instead of the value of 965.8 B. T. U. The difference in results due to this is but very slight.

71. *Equivalent evaporation from and at 212° per pound of combustible.* (*Item 61 divided by Item 30*) . . . . lb.
(If the equivalent evaporation, Items 69, 70, and 71, is not corrected for the quality of the steam, the fact should be stated.)

EFFICIENCY

(*Art. XXI, Code*)

72. *Efficiency of the boiler; heat absorbed by the boiler per pound of combustible divided by the heat value of 1 pound of combustible* per cent.
73. *Efficiency of boiler, including the grate; heat absorbed by the boiler, per pound of dry coal, divided by the heat value of 1 pound of dry coal* . . . . per cent.

COST OF EVAPORATION

74. *Cost of coal per ton of________pounds delivered in boiler room* . . . . $
75. Cost of fuel for evaporating 1,000 pounds of water under observed conditions . . . . $
76. *Cost of fuel used for evaporating 1,000 pounds of water from and at 212°* . . . . $

SMOKE OBSERVATIONS

77. Percentage of smoke as observed . . . . per cent.
78. Weight of soot per hour obtained from smoke meter . . . . oz.
79. Volume of soot per hour obtained from smoke meter . . . . cu. in.

METHODS OF FIRING

80. Kind of firing (spreading, alternate, or coking)
81. Average thickness of fire . . . .
82. Average intervals between firings for each furnace during time when fires are in normal condition . . . .
83. Average intervals between times of leveling or breaking up . . . .

ANALYSES OF THE DRY GASES

84. Carbon dioxide, CO_2 . . . . per cent.
85. Oxygen, O . . . . per cent.
86. Carbon monoxide, CO . . . . per cent.
87. Hydrogen and hydrocarbons . . . . . . per cent.
88. Nitrogen (by difference), N . . . . . . per cent.

100 per cent.

TABLE NO. 2

DATA AND RESULTS OF EVAPORATIVE TEST

ARRANGED IN ACCORDANCE WITH THE SHORT FORM ADVISED BY THE BOILER-TEST COMMITTEE OF THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS. CODE OF 1899.

Made by________ on________ boiler, at________
to determine____________________
Kind of fuel____________________
Kind of furnace____________________
Method of starting and stopping the test ("standard" or "alternate," Arts. X and XI, Code)__________

Grate surface . . . . sq. ft.
Water-heating surface . . . . sq. ft.
Superheating surface . . . . sq. ft.

TOTAL QUANTITIES

1. Date of trial____________________
2. Duration of trial . . . . hr.
3. Weight of coal as fired . . . . lb.
4. Percentage of moisture in coal . . . . per cent.
5. Total weight of dry coal consumed . . . . . . lb.
6. Total ash and refuse . . . . lb.
7. Percentage of ash and refuse in dry coal . . per cent.
8. Total weight of water fed to the boiler . . . . lb.
9. Water actually evaporated, corrected for moisture or superheat in steam . . . . lb.
10. Equivalent water evaporated into dry steam from and at 212° . . . . lb.

HOURLY QUANTITIES

11. Dry coal consumed per hour . . . . lb.
12. Dry coal per square foot of grate surface per hour lb.
13. Water evaporated per hour, corrected for quality of steam . . . . lb.
14. Equivalent evaporation per hour from and at 212° lb.
15. Equivalent evaporation per hour from and at 212° per square foot of water-heating surface lb.

AVERAGE PRESSURES, TEMPERATURES, ETC.

16. Steam pressure by gauge______pounds per square inch
17. Temperature of feedwater entering boiler . . deg.
18. Temperature of escaping gases from boiler . . deg.
19. Force of draft between damper and boiler . . . . in. of water
20. Percentage of moisture in steam, or number of degrees of superheating . . . . per cent. or deg.

HORSEPOWER

21. Horsepower developed (Item 14 divided by 34½) . . . . H. P.
22. Builders' rated horsepower . . . . H. P.
23. Percentage of builders' rated horsepower developed . . . . per cent.

ECONOMIC RESULTS

24. Water apparently evaporated under actual conditions per pound of coal as fired. (Item 8 divided by Item 3) . . . . lb.
25. Equivalent evaporation from and at 212° per pound of coal as fired. (Item 10 divided by Item 3) . . . . lb.
26. Equivalent evaporation from and at 212° per pound of dry coal. (Item 10 divided by Item 5) lb.
27. Equivalent evaporation from and at 212° per pound of combustible. [Item 10 divided by (Item 5 minus Item 6)] . . . . lb.

(If Items 25, 26, and 27 are not corrected for quality of steam, the fact should be stated.)

Efficiency

28.	Calorific value of dry coal per pound . .	B. T. U.
29.	Calorific value of the combustible per pound	B. T. U.
30.	Efficiency of boiler (based on combustible)	per cent.
31.	Efficiency of boiler, including grate (based on dry coal)	per cent.

Cost of Evaporation

32. Cost of coal per ton of____pounds delivered in boiler room . . . . $
33. Cost of coal required for evaporating 1,000 pounds of water from and at 212° . . . . $

WORKING UP THE DATA

74. The method of working up the data obtained during a boiler trial is shown by the following example:

Illustrative Example.—Given the following data taken during a boiler trial, required, to make the necessary calculations for economic evaporation and horsepower:

Duration of test	10 hours
Average gauge pressure	72 pounds
Average temperature of feedwater	122° F.
Pounds of coal burned	15,232 pounds
Percentage of ash	$4\frac{1}{2}$ per cent.
Water fed to boiler	124,600 pounds
Average quality of steam	97.2 per cent.
Rated horsepower	300

Calculations.—Water evaporated,

$$124{,}600 \times .972 = 121{,}111 \text{ pounds}$$

Water evaporated per pound of coal—actual conditions,

$$121{,}111 \div 15{,}232 = 7.95 \text{ pounds}$$

Water evaporated per pound of combustible,

$$121{,}111 \div \left[15{,}232 \times \frac{(100 - 4\frac{1}{2})}{100}\right] = 121{,}111 \div 14{,}546.7$$

$$= 8.33 \text{ pounds}$$

From Table I, the factor of evaporation for the given pressure and temperature of feedwater is 1.126. Hence, the equivalent evaporation from and at 212° per pound of coal is

$$7.95 \times 1.126 = 8.95 \text{ pounds}$$

The equivalent evaporation per pound of combustible is

$$8.33 \times 1.126 = 9.38 \text{ pounds}$$

The total equivalent evaporation from and at 212° F. per hour is

$$\frac{121{,}111 \times 1.126}{10} = 13{,}637.1 \text{ pounds}$$

The horsepower, therefore, is

$$13{,}637.1 \text{ pounds} \div 34\tfrac{1}{2} = 395.28 \text{ horsepower}$$

The per cent. above rated capacity is

$$\frac{(395.28 - 300) \times 100}{300} = 31.76 \text{ per cent.}$$

STEAM-BOILER DESIGN

(PART 1)

PROPORTIONS, POWER, AND CONSTRUCTION OF BOILERS

HEATING AND GRATE SURFACE

PROPORTIONS OF BOILER PARTS

1. The grate, heating surface, and steam space are the three parts of a steam boiler that enter most vitally into its capacity as a steam producer. A proper relation of these parts, giving each its correct size and position, will do much toward making a well-proportioned and efficient boiler. The location of the parts has been considered in *Types of Steam Boilers*, and the relative proportions of these parts may now be taken up.

HEATING SURFACE

2. The **heating surface** of a boiler includes the entire surface of the shell and flues coming in contact with the flame and furnace gases on one side and water on the other; this includes, in the case of externally fired boilers, the portion of the shell below the fire line, portions of the heads, and the inner surface of fire-tubes and flues, or the outer surface of water tubes. In horizontal return-tubular boilers, about one-half of the shell is usually exposed to the hot gases. In the case of internally fired boilers, the heating surface includes the interior of the firebox, or furnace flues, and the inner surface of the tubes, if there are any.

The heating surface of each head is equal to about two-thirds of its area minus the outside cross-sectional area of the tubes. Since the front head or tube plate is an inefficient heating surface, some authorities do not include it in calculating the effective heating surface.

3. The ratio of the heating surface to the grate area in each of the several types of boilers varies considerably in practice. The ratios usually adopted are as follows:

Type of Boiler	Ratio
Plain cylindrical	10 to 15
Cornish	15 to 40
Lancashire	25 to 33
Horizontal return tubular	25 to 50
Water tube	35 to 65
Marine fire-tube	25 to 40
Marine water tube	35 to 40
Locomotive	50 to 90

4. The designer must exercise judgment in assuming ratios that are suitable to the conditions under which the boilers are to be used. When the rate of combustion is high, a larger ratio must be employed than when the rate of combustion is low. From a large number of tests of horizontal tubular boilers, G. H. Barrus concluded that the ratio of heating surface to the grate area for that type of boiler should be 36 to 1 for anthracite coal, where the rate of combustion is not above 12 pounds per square foot of grate. For bituminous coal, the ratio may be from 45 to 50 to 1. He also found that with anthracite coal the highest efficiency is obtained when the combined cross-sectional area of the tubes is from one-ninth to one-tenth the grate area, and that with bituminous coal the greatest efficiency is obtained when the tube area is from one-fourth to one-seventh the grate area.

5. Since the greater part of the heating surface of tubular, and especially of locomotive, boilers is furnished by the tubes, particular attention should be paid to their arrangement.

The tubes are from $2\frac{1}{2}$ to 4 inches in diameter; their length should not exceed 5 feet per inch of diameter, and a shorter length would probably be an advantage. The tubes should be arranged in horizontal and vertical rows; if staggered, they hinder the circulation of the water. The pitch of the tubes—that is, the distance from center to center—should be from one and one-third to one and one-half times the diameter of the tube, and the distance between the two center rows should be double the distance between the other rows. The distance between the shell and the outer row of tubes should be 3 inches or more. It is good practice not to carry the tubes down near the bottom of the shell, as the lower tubes receive only the coolest gases, and therefore do not furnish efficient heating surface; by leaving them out, a large body of water rests directly over the fire and a good circulation is insured. The distance from the top of the tubes to top of the shell should be about two-fifths the diameter of the shell; if steam drums or domes are used, this height may be three-eighths or one-third the diameter of the shell.

6. The calculation of the heating surface of boiler tubes may be facilitated by referring to Table I, which gives dimensions of lap-welded boiler tubes.

EXAMPLE.—Calculate the heating surface of a boiler 5 feet in diameter, 15 feet long, and containing eighty-two 3-inch tubes. The boiler is so set that half the shell is exposed to the fire.

SOLUTION.—The heating surface of the shell is $\frac{1}{2} \times 5 \times 3.1416 \times 15 = 117.8$ sq. ft.

The heating surface of each head is two-thirds its area less the cross-sectional area of the tubes; the area of the tubes may be taken from Table I. Hence, the heating surface of heads is

$$2\left(\frac{2}{3} \times 5^2 \times .7854 - \frac{82 \times 7.0686}{144}\right) = 18.1 \text{ sq. ft.}$$

The total length of tubes is $82 \times 15 = 1{,}230$ ft., and from Table I the length per sq. ft. inside surface is 1.373 ft. Hence, the heating surface of tubes is $1{,}230 \div 1.373 = 896$ sq. ft.

The total heating surface is therefore $117.8 + 18.1 + 896 = 1{,}031.9$ sq. ft.

Leaving out the front head, the heating surface will be 1,022.85 sq. ft. Ans.

TABLE I

LAP-WELDED AMERICAN BOILER TUBES, STANDARD DIMENSIONS

Diameter Inches		Thickness		Circumference Inches		Transverse Areas Square Inches			Length of Tube Feet per Square Foot of Surface		Nominal Weight per Foot Pounds
Outside	Inside	Inches	Nearest B.W.G.*	Outside	Inside	Outside	Inside	Metal	Outside	Inside	
1	.810	.095	13	3.142	2.545	.7854	.5153	.2701	3.819	4.715	.90
1½	1.310	.095	13	4.712	4.115	1.7671	1.3478	.4193	2.547	2.916	1.40
2	1.810	.095	13	6.283	5.686	3.1416	2.5730	.5686	1.909	2.110	1.91
2½	2.282	.109	12	7.854	7.169	4.9087	4.0899	.8188	1.528	1.674	2.75
2¾	2.532	.109	12	8.639	7.954	5.9396	5.0349	.9047	1.389	1.508	3.04
3	2.782	.109	12	9.425	8.740	7.0686	6.0787	.9899	1.273	1.373	3.33
3¼	3.010	.120	11	10.210	9.456	8.2958	7.1157	1.1801	1.175	1.269	3.96
3½	3.260	.120	11	10.996	10.242	9.6211	8.3469	1.274	1.091	1.171	4.28
3¾	3.510	.120	11	11.781	11.027	11.045	9.6762	1.369	1.018	1.088	4.60
4	3.732	.134	10	12.566	11.724	12.566	10.939	1.627	.955	1.024	5.47
4½	4.232	.134	10	14.137	13.295	15.904	14.066	1.838	.849	.902	6.17
5	4.704	.148	9	15.708	14.778	19.635	17.379	2.256	.764	.812	7.58
6	5.670	.165	8	18.850	17.813	28.274	25.249	3.025	.637	.673	10.16
7	6.670	.165	8	21.991	20.954	38.485	34.941	3.544	.546	.573	11.90
8	7.670	.165	8	25.133	24.096	50.265	46.204	4.061	.477	.498	13.65
9	8.640	.180	7	28.274	27.143	63.617	58.629	4.988	.424	.442	16.76
10	9.594	.203	6	31.416	30.140	78.540	72.291	6.249	.382	.398	21.00

* Birmingham wire gauge.

GRATE SURFACE

7. The **grate surface,** or **grate area,** of a steam boiler depends on the rate of combustion, the evaporation per pound of fuel, and the total weight of steam evaporated per hour. Suppose that a boiler generates 18,000 pounds of steam per hour, that the rate of combustion is 16 pounds of coal per square foot of grate per hour, and that the evaporation is 9 pounds of water per pound of coal. Then, the necessary grate surface must be $\frac{18{,}000}{16 \times 9} = 125$ square feet. This would ordinarily be divided among several furnaces, as a grate much longer than 6 feet cannot easily be fired, while it is usually thought advisable to make the width not greater than from 4 to 6 feet. Assuming the dimensions of the grate to be 4 ft. × 6 ft., the grate area just mentioned would require $\frac{125}{4 \times 6} = 5$ furnaces.

From this method of calculating grate area, the following formula may be readily deduced:

Let G = area of grate, in square feet;
F = rate of combustion, in pounds per square foot of grate surface per hour;
W = weight of steam generated per hour by the boiler or boilers;
e = evaporation, in pounds of water per pound of coal.

Then,
$$G = \frac{W}{Fe}$$

In using this formula, W and e should be taken at the same pressure and temperature. That is, if the boiler generates W pounds of steam per hour at a pressure of 80 pounds from a temperature of 60°, e must represent the number of pounds of water that a pound of coal will raise from 60° and evaporate into steam at 80 pounds pressure. For the purpose of calculation, it is customary to reduce both W and e to the equivalent evaporation from and at 212° F.

EXAMPLE.—Find the grate area of a boiler that evaporates 2,400 pounds of water from and at 212° per hour, the rate of combustion being 12 pounds per square foot of grate surface per hour, and the evaporation $10\frac{1}{2}$ pounds of water from and at 212° per pound of coal.

SOLUTION.—Using the formula,

$$G = \frac{2,400}{12 \times 10.5} = 19 \text{ sq. ft.} \quad \text{Ans.}$$

8. Steam Space.—The **steam space** required by a given boiler depends on the purpose for which the steam is required. Where the steam is under high pressure, and where relatively small quantities are withdrawn at very frequent intervals, as in locomotives, the steam space need not be so great as where large quantities are withdrawn less frequently. Where the boiler supplies steam to an engine, the steam space, as a general rule, should hold enough steam to supply the engine for from 20 to 25 seconds. When steam is supplied for heating purposes, it need not necessarily be dry, and, hence, the steam space may be smaller. As ordinarily designed, from one-quarter to one-third the cubic contents of the boiler is steam space and the remainder water space.

9. Proportions of Boilers.—The proportions of horizontal tubular and vertical boilers adopted by a leading manufacturing company are given in Tables II and III.

EXAMPLES FOR PRACTICE

1. Find the grate area of a boiler that evaporates 1,800 pounds of water from and at 212° per hour, the rate of combustion being 10 pounds of coal per square foot of grate surface per hour, and the evaporation 9 pounds of water from and at 212° per pound of coal.
Ans. 20 sq. ft.

2. Calculate the heating surface of a return-tubular boiler 54 inches in diameter, 16 feet long, and containing sixty-two 3-inch tubes. The boiler is so set that half of the shell is exposed to the fire, and the front head is not considered. Ans. 843.16 sq. ft.

TABLE II

PROPORTIONS OF VERTICAL BOILERS

Horse-power	Diameter of Shell Inches	Height of Boiler		Diameter of Tubes Inches	Length of Tubes		Number of Tubes	Heating Surface Square Feet	Diameter of Grate Inches	Total Grate Area Square Feet	Total Tube Area Square Feet
		Feet	Inches		Feet	Inches					
34	48	10	8	2½	7	8	91	340	42½	9.722	2.584
28	44	11	4½	2½	8	5¼	64	282	38	8.08	1.8176
24	42	9	5½	2	6	7⅜	96	247	36	7.068	1.6992
23	40	9	8½	2	6	11¼	85	233	34¼	6.398	1.5045
20	38	8	9¼	2	6	1	85	201	32½	5.760	1.5045
17	36	8	8½	2	6	1¼	73	177	30½	5.073	1.2921
15	34	8	7¼	2	6	1½	61	152	28¼	4.352	1.0797
12	32	7	6¼	2	5	1	61	124	26½	3.83	1.0797
11	30	7	6¾	2	5	2¼	55	117	24½	3.343	0.9735
10	28	7	8½	2	5	5	42	100	23	2.885	0.7434
8	26	7	4¼	2	5	2	37	81	21	2.335	0.6549
6.8	24	7	10¾	2	5	9½	26	68	19	1.967	0.4602
5	22	7	4¼	2	5	4	22	53	17	1.527	0.3894
4	20	8	3½	2	6	4½	14	42	15	1.227	0.2478

TABLE III

PROPORTIONS OF HORIZONTAL RETURN-TUBULAR BOILERS

Horse-power	Diameter Inches	Length Feet	Diameter of Tubes Inches	Number of Tubes	Heating Surface of Tubes Square Feet	Heating Surface of Shell Square Feet	Total Heating Surface Square Feet	Grate Area Square Feet	Length of Grate		Width of Grate	
									Feet	Inches	Feet	Inches
120	72	20	3½	95	1,622.6	202.9	1,825.5	52	8		6	6
110	72	18	3½	95	1,460.3	184.1	1,644.4	47	8		6	
100	66	18	3	102	1,336.6	169.3	1,505.9	43	7	2	6	
90	66	16	3	102	1,188	150	1,338	38	6	4	6	
80	60	16	3	86	1,001.7	135.1	1,136.8	32	6		5	3
70	60	15	3	82	895.4	127.4	1,022.8	29.2	5	6	5	3
60	54	16	3	62	722.1	121.2	843.3	25.6	5		5	1½
50	54	15	3	56	611.5	114.5	726	20.7	4	7	4	6
40	48	14	3	44	451.7	94.6	546.3	15.6	3	11	4	
30	48	12	3	40	349.4	82.3	431.7	12.3	3	2	4	
20	40	10	3	28	205.8	57.3	263.1	7.5	2	2	3	6

HORSEPOWER OF BOILERS

10. The **horsepower** of a boiler is a measure of its capacity for generating steam. Boilermakers usually rate the horsepower of boilers as a certain fraction of the heating surface. Thus, in Tables II and III it will be noticed that for the vertical boilers the horsepower is one-tenth the heating surface, and for the horizontal type it is about one-fifteenth the heating surface. The ratio of heating surface to horsepower, when the boiler is run under ordinary conditions, is about as follows:

Type of Boiler	Square Feet of Heating Surface per Horsepower
Water tube	10 to 12
Return tubular	12 to 16
Flue	8 to 12
Plain cylindrical	6 to 10
Locomotive (stationary practice)	12 to 14
Vertical	11 to 20

The foregoing method of rating boilers is extremely indefinite; with the same heating surface, different boilers of the same type may, under different circumstances, generate very different quantities of steam.

BOILER MATERIALS AND DETAILS OF CONSTRUCTION

PROPERTIES OF MATERIALS

GENERAL REQUIREMENTS

11. The materials used in the construction of steam boilers are mild steel, steel castings, wrought iron, cast iron, copper, and brass. The steel and the wrought iron must be ductile, in order that they may withstand the bending, punching, drilling, and riveting to which they will be subjected. Their tensile strengths must be high and they must also be tough and elastic. Other materials used should have these same qualities as far as possible, and all castings should be free from the flaws and weak places that so often occur.

STEEL AND WROUGHT IRON

12. Steel.—Steel is now generally used for all parts of a boiler, although wrought iron also has been used for stays and fastenings. Steel suitable for boiler plates may have an ultimate tensile strength of from 55,000 pounds to not more than 65,000 pounds per square inch, and an elastic limit of from 30,000 to 31,000 pounds per square inch, with an elongation of 25 per cent. in 8 inches and 50 per cent. reduction of area at the point of rupture. The plates should be free from sulphur and phosphorus, as sulphur makes the steel brittle when hot and phosphorus makes it brittle when cold. There should not be more than .04 per cent. of either phosphorus or sulphur in the steel. When cold or when heated to a bright red and quenched in water of about 80° temperature, good boiler steel should stand bending double without sign of fracture.

13. Firebox and Flange Steel.—There are two kinds of steel used for boiler plates, namely, **firebox steel** and **flange steel;** the latter is more ductile than the former and better suited to withstand flanging. Firebox steel successfully resists the severe straining actions resulting from contact with the fire and from sudden changes in temperature when firing. There is no difference, however, in the chemical compositions of the two grades of steel. The greater ductility in one is due to physical conditions that result from greater care in the manufacture, which should be by the open-hearth process. All flanging and bending of steel should be done at or above a low red heat, and when flanging by hand, wooden hammers or mallets only should be employed. Steel at a temperature known as blue heat will crack if bent, while at a red heat or when cold, it will permit bending or flanging without injury; consequently, steel should never be worked at a blue heat.

14. Rivets.—Rivets are made either of wrought iron or steel. Wrought-iron rivets should have a tensile strength of 48,000 pounds per square inch, and steel rivets about 55,000 pounds. Such properties as elastic limit, elongation, and reduction of area should be the same as for the plates. A rivet should bend around a bar of its own diameter without fracture when cold, and should bend double without fracture when hot.

15. Wrought Iron.—**Wrought iron** is freer working than steel, is less liable to injury under rough handling, and may be welded. Wrought-iron plates of a quality suitable for boiler shells are more expensive than steel plates of equal strength, but may be flanged and scarfed (the edges beveled) without injury at a lower temperature than steel, thus saving reheating. Wrought-iron stays, rods, etc. should have an ultimate tensile strength of 48,000 pounds per square inch, while plates suitable for boiler shells should have a tensile strength of 45,000 pounds per square inch and an elastic limit of 23,000 pounds per square inch.

CAST IRON

16. Cast iron is not much used in boiler construction, except for supporting lugs or brackets and for manhole frames and covers. Even for the latter purpose, either forged steel or steel casting is better. Cast iron should never be used where it will be subject to sudden changes of temperature or to unequal expansion. It has a tensile strength of about 20,000 pounds per square inch and a compressive resistance of about 90,000 pounds per square inch.

COPPER AND COMPOSITION METALS

17. Copper was at one time much used for fireboxes of locomotives, but it is now too expensive except for such special cases as torpedo-boat and fire-engine boilers. It has a tensile strength of 34,000 pounds per square inch, and from 20 to 25 per cent. of elongation in 8 inches. On account of its greater ductility and thermal conductivity, copper plates may be much thicker than those made of iron. Copper is also used for making steam pipes, being rolled from plates and scarfed and brazed at the joint; the pipe is then brazed into gun-metal or composition flanges. The serious accidents caused in recent years by the rupture of a number of these pipes through the high steam pressures used has created a strong inclination to employ only wrought-iron or steel pipe for steam. The various alloys, known as brass, bronze, and composition, are used for fittings and trimmings, valve seats, and parts subject to corrosion and hard usage.

TESTING OF BOILER MATERIALS

METHODS OF TESTING

18. The suitability of a given piece of material for boiler construction is determined by means of a number of tests, some of which are chemical and some physical. Chemical tests, however, are seldom used except by the manufacturer of the material. The most important of the physical tests is that for *tensile strength*, both because of the direct relation it bears to the actual conditions under which the material is used in the boiler and because of the accuracy with which the test can be made. The *bending test* is also important for boiler material, and is used quite extensively. Rivets are sometimes subjected to a shearing and a bending test, to find their strength in shear and in bending.

19. Test for Tensile Strength.—Strips are cut from the plates as they come from the rolls, or from the rolled rivet rods, for the specimens on which tensile tests are to be made. In order to secure uniform and reliable results, it is important to have the specimens carefully prepared.

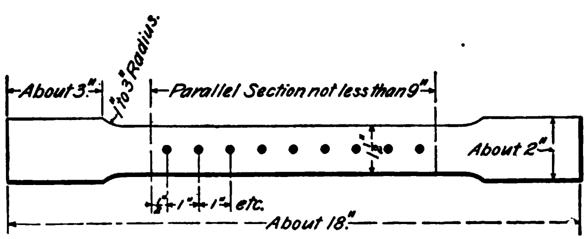


FIG. 1

Standard sizes of test pieces have been adopted for this purpose, and for boiler plate they should be 1½ inches wide, planed parallel on both edges and with the parallel section not less than 9 inches long between shoulders, as shown in Fig. 1. This form has been adopted as the manufacturers' standard specimen. Along its length for a distance of

8 inches, the specimen is divided into spaces 1 inch long, the division being made by very light center-punch marks; this is done in order that the amount of greatest increase in length may be located and the increase for each inch measured. The tensile strength of the specimen is generally determined by means of a testing machine that will register automatically, in pounds, the force exerted on the test piece in order to pull it apart. A device is also sometimes provided to measure the amount the piece is stretched.

The United States Navy standard tensile test piece for materials thick enough to be turned is shown in Fig. 2, the dimensions being given in inches. It is also desirable to test full-sized staybolts and stay braces. The data obtained by means of the testing machine are the ultimate tensile

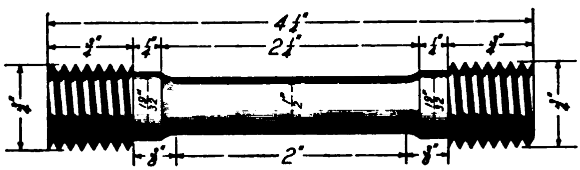


FIG. 2

strength of the specimen; maximum stress that the specimen will sustain without permanent set; elongation, or the total amount the specimen has stretched; and the reduction of area of the specimen at the point of rupture.

20. Shearing Test.—As the rivets of a boiler shell are subject to a shearing stress, they are sometimes tested by riveting two plates together with the rivets to be tested and then pulling the plates apart in a tension machine and shearing off the rivets. Such a test gives the ultimate shearing strength of the rivets. Numerous tests have shown the average shearing resistance of iron rivets in single shear to be 38,000 pounds per square inch, and of steel rivets, 45,000 pounds per square inch. The rod from which the rivets are made generally has a shearing strength of about two-thirds the tensile strength.

From tests conducted by the Master Steam Boiler Makers Association, the values for the shearing strength of rivets, in pounds per square inch, were found to be as follows: 42,000 for iron rivets in single shear; 80,000 for iron rivets in double shear; 45,000 for steel rivets in single shear; and 88,000 for steel rivets in double shear. In practice there is considerable variation in the values used, as there is no generally accepted standard.

21. Bending Test.—The **bending test** for steel boiler plate or rivets consists in bending a specimen double and closing it down tight. Two bending tests are often required. In one, the specimen is bent cold without any treatment, and, in the other, the specimen is heated to a bright red and then quenched, or suddenly cooled, in water having a temperature of about 80° F. The former is called a *cold bend* and the latter a *quench bend.* A good quality of steel for boiler plates or rivets should bend flat on itself, either cold or quenched, without any indication of cracking on the outside of the bend.

COUPONS

22. In some cases, the plates to be used for boilermaking are not sheared at the mills to the exact size required, but are furnished to the boilermaker with a strip that can be cut from the plate and tested at the boiler shop. In this way, the user of the plate can satisfy himself that the quality of the plate conforms to the specifications. The strip that is to be cut from the plate for testing is called a **coupon.**

STAMPING

23. Many specifications require boiler plates to be stamped with the name of the maker of the plate and with its tensile strength in pounds per square inch. This stamp should be so placed on a part of the sheet that it can be readily seen after the boiler is made. The United States laws governing the construction and inspection of marine boilers specify that every iron or steel plate intended for the construction of boilers to be used on steam vessels shall be stamped by the manufacturer in the following manner: "At the corners, at a distance of about 8 inches from the edges, and at or near the center of the plate, with the name of the manufacturer, the place where manufactured, and the number of pounds tensile stress it will bear to the sectional square inch."

DETAILS OF CONSTRUCTION

RIVETED JOINTS

24. Riveting is one of the most important operations in boilermaking, for, in a large measure, the strength and safety of the boiler depend on the character of the riveting. The flat plates, as received from the rolling mill, are first sheared or cut to the proper size, and the rough edges are planed off. The rivet holes are then punched or drilled in the edge of the plate, and the plate is bent to the required shape, after which it is ready for riveting. Drilling is more expensive but less injurious than punching. The metal nearest the holes is damaged by the punching process, but the injury does not extend far from the holes. It is quite common practice to punch the holes small and ream them to size in place, thus removing the metal damaged by the punching.

25. Arrangement of Joints and Plates.—The plates of externally fired boilers should be so arranged that the riveted joints are as far as possible from the fire. This may be accomplished by using extra-large plates for the furnace end of the shell. In fact, it is customary to build horizontal return-tubular boilers with only one horizontal seam to the section. The seam is then placed above the fire line and in such a position as not to interfere with the dome or the wall brackets. In order to accomplish this, it is customary to locate the longitudinal seam at about 45° from the horizontal center line, as shown at *a* and *a'*, Fig. 3. The dome may then be placed as shown at *b* and the brackets as shown at *c*, without interfering with the joint.

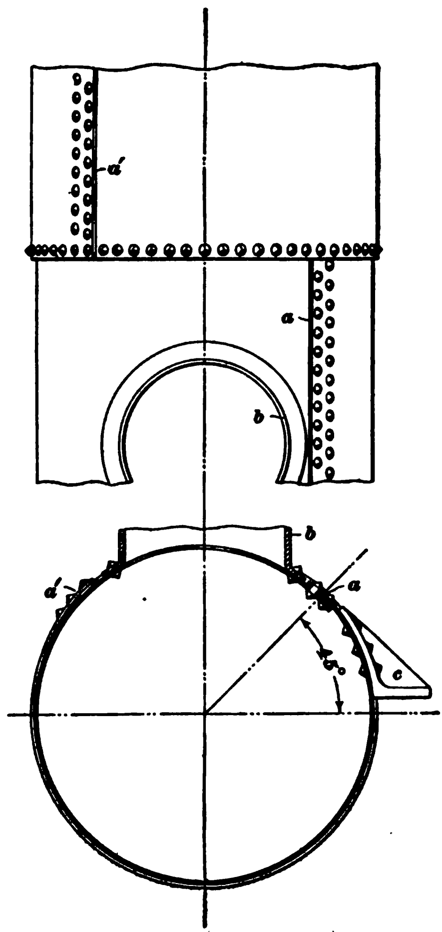


Fig. 3

When a cylindrical shell consists of two or more sections, or courses, the joints should be arranged to alternate, or break, as shown at *a* and *a'*. Riveted joints of internally fired cylindrical boiler flues, or furnaces, are generally located below the grates and as far to one side as possible. Thus, by being situated below the grates, they are away from the fire, and, by being at one side, they do not interfere with the removal of the ashes.

In order to make a tight joint where three thicknesses of plates are used, as at the point where longitudinal and girth seams come together, the inner plate of the longitudinal seam must be scarfed, or hammered thin at the edges. The arrangement of the rivets in the longitudinal seam adjoining the girth seam requires some care. In a double-riveted lap joint, shown in Fig. 4, and in a double-riveted butt joint, shown in Fig. 5, the pitch of the rivets should be uniform in the outer row *a*, thus creating an unequal spacing at the end of the inner row *b*. In a single-riveted lap joint, shown in Fig. 6, it is frequently necessary to make the spacing at *a*, next to the girth joint, somewhat greater than the regular pitch of the rivets. This is done to give sufficient clearance to the rivet dies, so that they will not cut the inside lap when driving the rivet.

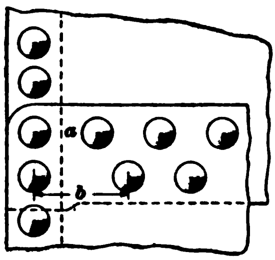


FIG. 4

In the construction of both vertical and horizontal shells, the inside lap is usually placed so as to face *downwards*, for if it faces upwards, a ledge is formed on which sediment may be deposited. Since wrought-iron plates are stronger in the direction of the fiber, they should be arranged so that the fiber runs circumferentially around the shell; that is, in the direction of the girth seams.

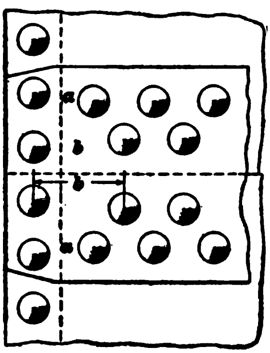


FIG. 5

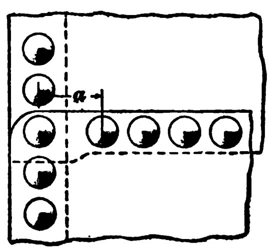


FIG. 6

26. Methods of Connecting Plates.—Different methods of connecting plates at right angles are shown

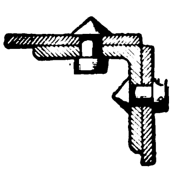


FIG. 7

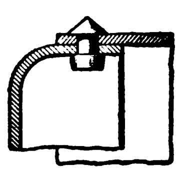


FIG. 8

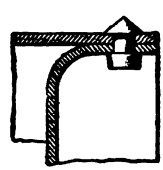


FIG. 9

in Figs. 7 to 14. In Fig. 7, the two plates are riveted to an angle iron. This construction is used sometimes for connecting the heads of a boiler to the shell. Figs. 8 and 9 show the head flanged and riveted to the shell. Iron for flanging should be of the best quality. The radius of the curve to which the head is flanged should be at least four times the thickness of the plate.

27. In Figs. 10 to 14 is shown the usual construction of the water legs and furnace doors of vertical and firebox

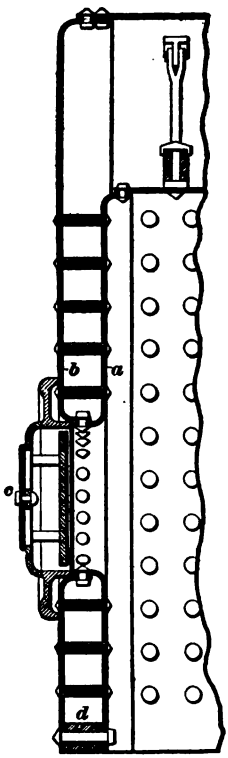


Fig. 10

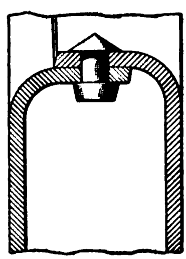


Fig. 11

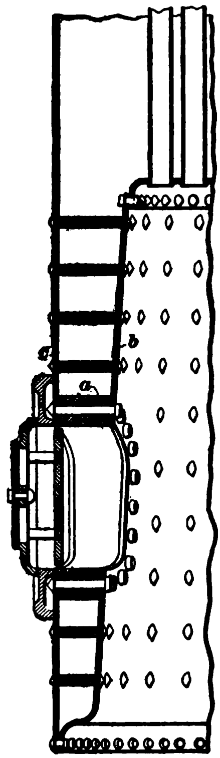


Fig. 12

boilers. Fig. 10 shows the door constructed by flanging the furnace sheet *a* and the front sheet *b* of the boiler. Single riveting is shown here, although the joint is frequently double-riveted. An enlarged view of this construction is shown in Fig. 11. The door *c*, Fig. 10, is generally made of cast iron, and is hinged to a cast-iron frame, which is usually held in position by four studs $\frac{7}{8}$ inch in diameter. Sometimes, the frame is omitted and the door is made of wrought iron; it is then held in position by riveting the hinges to the boiler. Around the lower ends of the water legs, or around the bottom of the furnace, and between the inside and outside furnace plates is riveted a wrought-iron ring *d*. In cheap boilers, this ring is frequently made of cast iron.

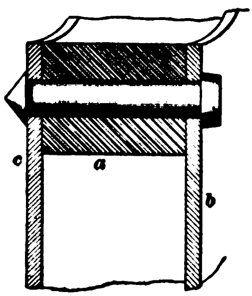


FIG. 13

28. In Fig. 12 is shown another method of constructing the opening for the furnace door and the bottom of the water leg. In this construction, the wrought-iron ring *a* is placed between the furnace plate *b* and the shell *c* of the boiler and riveted to them. An enlarged view of this construction is shown in Fig. 13. At the bottom of the water leg, the furnace plate is flanged and riveted to the shell as shown.

In the construction of the door opening through the water legs, as shown in Fig. 11, the rivets in the water leg are difficult to put in, and the construction shown in Fig. 13 is not generally considered desirable on account of the excessive thickness of the metal at the joint. Fig. 14, however, shows a joint between the furnace plate *a* and boiler shell *b* around the door that is now frequently used to overcome the objections mentioned. Both the firebox sheet and the water-leg sheet are flanged outwards and riveted as shown. This construction requires the best grade of material for the firebox sheet, and great care should be taken that this sheet is not drawn too thin at the outer edge.

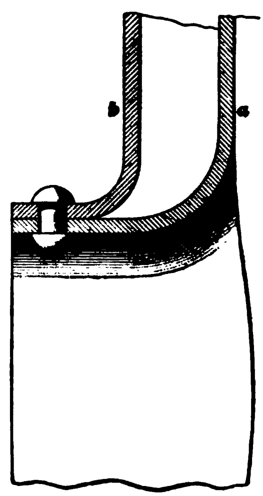


FIG. 14

BOILER STAYS

29. In general, the surfaces of boiler shells are either cylindrical, hemispherical, or flat. A cylinder or sphere subjected to an internal steam pressure is self-supporting; that is, the steam pressure tends to maintain the cylindrical or spherical form of the vessel and hinders distortion instead of producing it. If, on the contrary, the vessel is composed of flat surfaces, the steam pressure tends to distort it, giving it an approximately spherical form. Hence, flat surfaces, unless extremely thick, cannot be self-supporting, and must be braced, or stayed, to prevent distortion.

In the most common types of boilers, the flat surfaces that require staying are the upper sections of the flat heads of return-tubular boilers, the flat sides and crown sheets of the firebox in boilers of the locomotive type, and the combustion chambers of marine boilers.

30. In cylindrical tubular boilers, the flat ends above the tubes and around manholes are the only parts that require special staying. The tubes expanded into the heads below the water-line give sufficient support to the lower parts of the heads. The holding power of expanded tubes is from 15,000 to 22,000 pounds per tube, and in some cases much greater. Frequently, the tube will pull apart before it will draw out of the tube-sheet. When a manhole is placed in the bottom of the tube-sheet, the flat surface thus exposed must be stayed. Flat surfaces not exposed to a very high pressure are sometimes stayed by simply riveting **T** irons or channel irons to the surface. This method, however, is unsatisfactory and should not be used.

31. **Stayrods.**—In return-tubular and some other boilers, the flat heads are stayed by rods connecting the two heads. These rods, known as **direct stays**, are all designed in the same manner in regard to strength, but the method of fastening varies somewhat. In Fig. 15, the rod shown at *a* is upset at the end *b* for the threading, which is long enough to pass through the shell. A channel iron *c* is riveted to the shell and allows nuts and washers to be put on both inside and outside. On the outside, a copper washer *d* is placed next to the shell and a wrought-iron cap nut *e* is screwed over the end of the rod. This combination is very successful in preventing leakage around the end of the rod. A cast-iron washer *f* and a wrought-iron nut *g* are used on the inside.

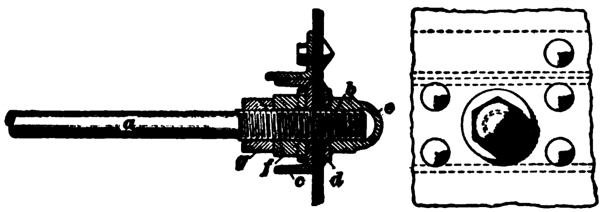


FIG. 15

Instead of the cap nuts on the ends of the rods, the plain form of nut is often used on both sides of the plate, as shown at *a*, *a′*, Fig. 16. Sometimes, the washers *b*, *b′* are large and riveted to the head-plate, thus very considerably increasing the area supported by the stay. The hole through which the rod passes is made large enough to clear the threads, and the space around the threads is filled with asbestos packing, as shown at *c*.

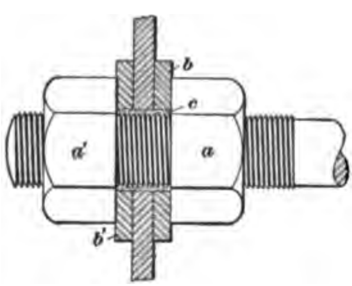


FIG. 16

32. Another method of connecting the ends of stayrods to the plate is shown in Fig. 17. Here, the ends of the stayrod *a*, instead of being threaded, are forked to receive the V-shaped connection *b*, which is riveted to the head of the boiler and joined to the stayrod by a bolt *c*. The combined effective area of the two legs of the rod *a* should exceed the area of the cross-section of the solid rod.

33. Staybolts.—Where the flat surfaces of the boilers come close together, short connecting bolts are necessary; these are distinguished from stayrods by being called

staybolts. The method of fastening staybolts is quite different from that required for stayrods, because of the

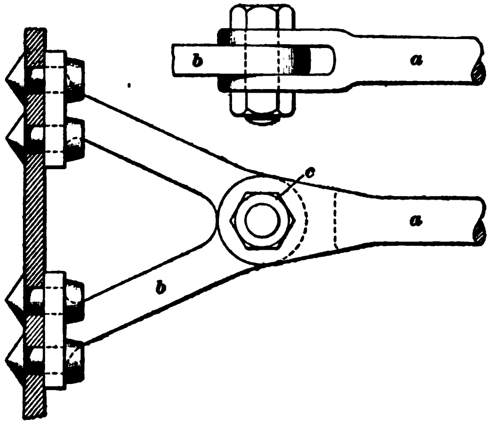


FIG. 17

difficulty of putting nuts on the bolt inside of the parts to be supported. Staybolts are therefore screwed into both plates.

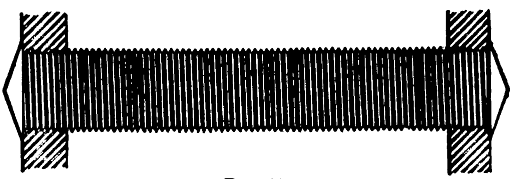


FIG. 18

Figs. 18, 19, and 20 illustrate different forms of screw staybolts used for strengthening the flat side surfaces, and

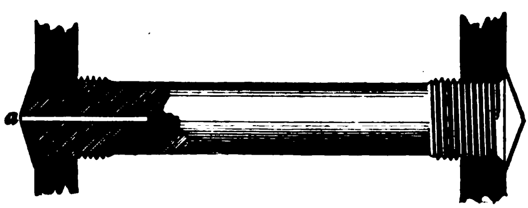


FIG. 19

in some instances the curved surfaces, of boilers. The one shown in Fig. 18 is generally used for staying the flat side surfaces of firebox boilers and the circular furnace sheets of vertical boilers. These bolts have an outside diameter of from about $\frac{7}{8}$ inch to about $1\frac{1}{8}$ inches, and are screwed into the plates, the ends being riveted over so as to give additional strength. The staybolt shown in Fig. 19 is used for

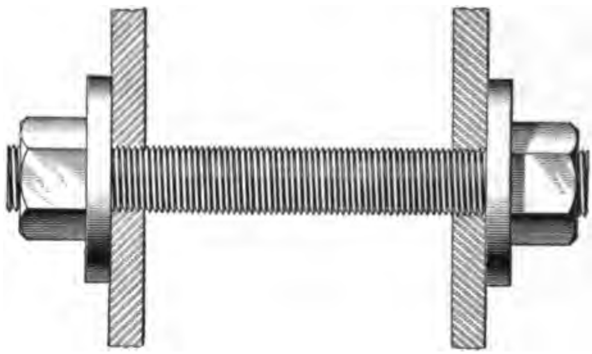


FIG. 20

the same purpose, but here the bolt is threaded only at the ends, which are upset. A small hole *a* is drilled into the axis of the bolt from each end for the purpose of detecting broken bolts; most cracks in staybolts occur near the plates. When a staybolt of this type is cracked, steam will issue from the hole.

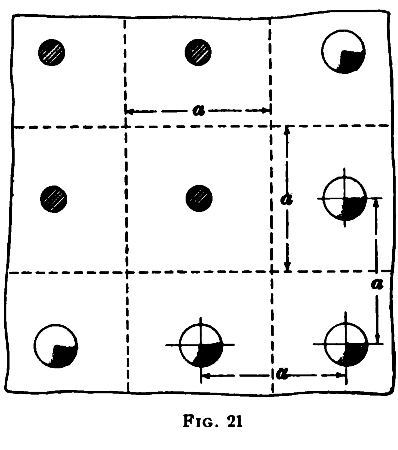


FIG. 21

34. The staybolt shown in Fig. 20 is generally used for staying the flat side surfaces of marine boilers. Instead of riveting the ends, washers and nuts are used, the washers serving to strengthen the flat plates by distributing the support. When these plates are of sufficient strength, the washers are dispensed with, and the nuts are screwed close to the plates. The staybolts used in marine boilers are made of steel, and vary in diameter from about $1\frac{1}{4}$ to $1\frac{5}{8}$ inches.

Firebox staybolts are generally arranged with the same pitch, both vertically and horizontally, as shown in Fig. 21. Denoting this pitch by a, it is evident that each staybolt supports an area a^2 represented by the dotted lines, the side a of the square being equal to the distance between the rivets. In some cases, however, the pitches of the stays, both vertically and horizontally, are not the same, and the area of the stay is then generally made great enough to support an area equal to one-half the sum of the squares of the two pitches. Thus, if the pitches are 6 and 9 inches, the stay is made strong enough to support an area equal to $(6 \times 6 + 9 \times 9) \div 2 = 58\frac{1}{2}$ square inches. It will be seen that this area is somewhat greater than the product of the two pitches, which would be the actual area supported by each stay.

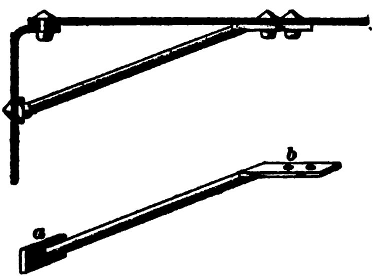


FIG. 22

35. Diagonal Stays.—When the boiler shell is long in proportion to its diameter, through stayrods are seldom used owing to the great lengths required; therefore, diagonal and gusset stays have been adopted. The **diagonal, or crowfoot, stay** is shown in Fig. 22. The end a is either bolted or riveted to the boiler head, and the end b is riveted to the shell.

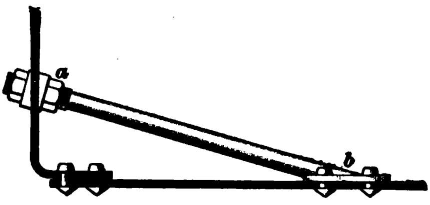


FIG. 23

In marine boilers, where this style of stayrod is used, instead of the end a being riveted to the head, it is threaded and supplied with nuts and taper washers on each side of the head, as shown in Fig. 23, the washers having such a taper that when one of the faces is against the head, the other is parallel with the faces of the nuts.

The hole for the stayrod is not threaded, but is made sufficiently large to allow the stay to pass through. The end *b* is riveted to the shell. Many engineers consider this form of brace undesirable, owing to the difficulty of fitting the brace tightly to the head.

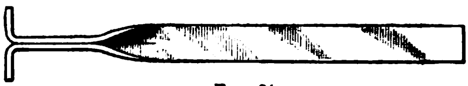


FIG. 24

36. One form of crowfoot brace is made of steel in one piece and without welds, as shown in Fig. 24. The design of the crowfoot should be such that the rivets are as close together as possible and are so located in relation to the center line of the brace as to be subjected to a direct pull only. If the rivets are not in the center line of the brace, additional stress is thrown on them. Unless the crowfoot is made extra stiff, or if the rivets are spaced very far apart, as shown in Fig. 25, the rivets will be subjected to a very great stress on account of the leverage in the crowfoot. A better arrangement of the rivets is shown in Fig. 26. The combined area of the rivets attaching the stay to the shell should be about equal to the area of the rod. The angle that a diagonal stay makes with the shell should not exceed 30°, and should be as much smaller as possible.

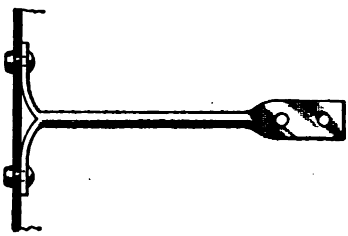


FIG. 25

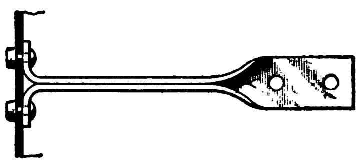


FIG. 26

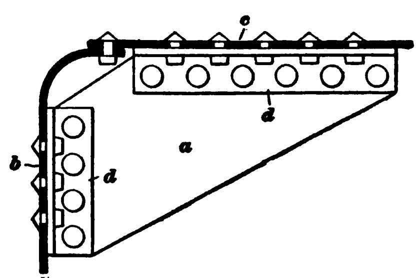


FIG. 27

37. Gusset Stays.—In Fig. 27 is shown a **gusset stay**, which consists of a wrought-iron or steel plate *a* secured to the boiler head *b* and shell *c* by either angle or **T** irons, as

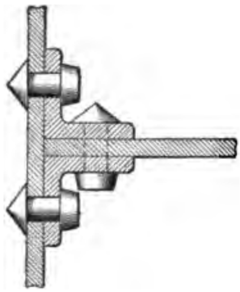


FIG. 28

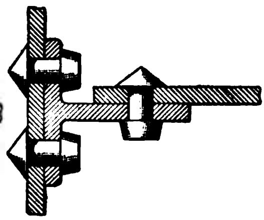


FIG. 29

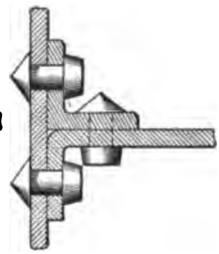


FIG. 30

shown at *d*, *d*. This form of stay is much used for staying the heads of internally fired boilers of the Lancashire and Galloway type.

In Figs. 28, 29, and 30 are shown three methods of fastening gusset stays. Fig. 28 shows a method of fastening the plate by using two angle irons with the plate between them, Fig. 29, a method of using a **T** iron with the plate riveted to one side and Fig. 30, a method in which the gusset plate is flanged to one side and the angle iron riveted to the other. Gusset stays are placed radially in a boiler, the largest one in the center and smaller ones to the right and left of it, as shown in Fig. 31.

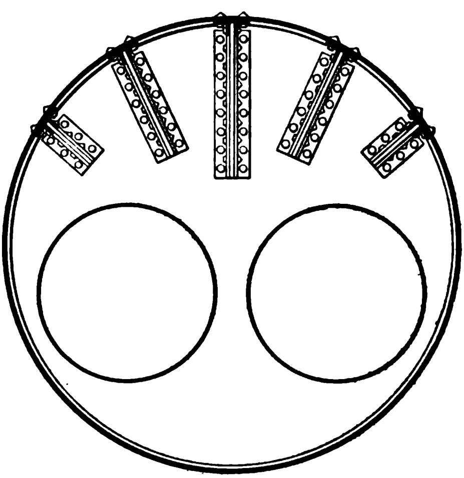


FIG. 31

Compared with direct or diagonal stays, gusset stays have the advantage of simplicity, and interfere but little with the accessibility of the boiler. Their rigidity, however, is objectionable because it localizes the strains and is liable to cause grooving. It is customary to calculate the size of gusset stays on the assumption that the resultant stresses act along a line midway between the edges not riveted, and that they are evenly distributed over the section at right angles to that line. Since the method is only approximate, a large factor of safety should be assumed.

38. Girder Stays, or Crown Bars.—It was previously stated that the side furnace plates of firebox boilers, the side plates of combustion chambers in marine boilers, and

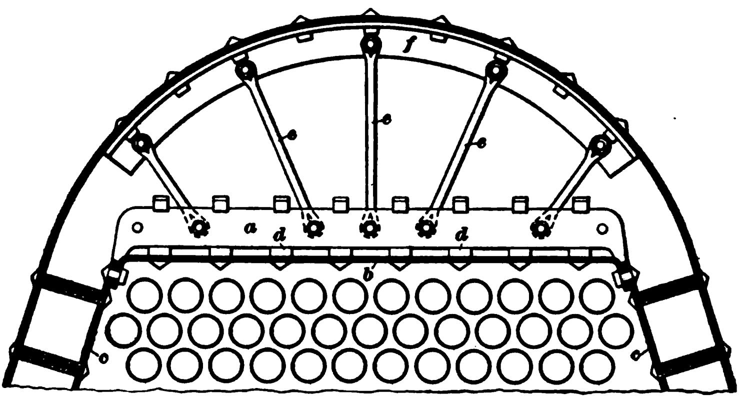


FIG. 32

the circular furnace plates of vertical boilers were strengthened with staybolts, but as yet nothing has been said regarding the staying of the upper plates or crown sheets. For strengthening these sheets in firebox and marine boilers, **girder stays, or crown bars**, are used; in vertical boilers, no special means are applied, since the tubes themselves furnish the required staying.

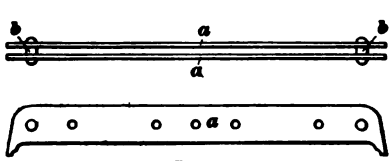
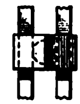


FIG. 33

The general staying of the crown sheet of a firebox boiler is shown in Fig. 32, in which *a* is a girder stay, or crown bar, resting partly on the crown sheet *b* and partly on the side furnace plates *c*, *c*, to which the crown sheet is riveted. Distance pieces *d*, *d* are used to prevent the crown sheet from bulging upwards when upsetting the rivets. Fig. 33 shows the general construction of the crown bars. They consist of two wrought-iron bars *a*, *a*, held apart at both ends by distance pieces *b*, *b* and are riveted together. The rivets used for staying the crown sheets of firebox boilers to the crown bars are of wrought iron, and are generally made like the one shown in Fig. 34. The projections *a*, *a* on the head of the rivet are to prevent the bars from spreading.



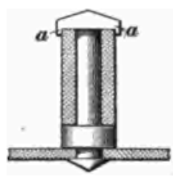


FIG. 34

In staying the crown sheets of the combustion chambers of marine boilers, bolts are used instead of rivets. The bolts are made of steel and are provided with nuts on each end, as shown at *a*, *a'*, Fig. 35, the washer *b* with lugs *c*, *c* being used to prevent the bars from spreading. In fire-box boilers, the crown sheet is further strengthened by stayrods *e*, as shown in Fig. 32, one of the ends of the rod being secured to the crown bar by pins, and the other to a **T** iron *f* that is riveted to the shell of the boiler. These rods not only give additional strength to the crown sheet, but also brace the cylindrical part of the shell. Rivets used for staying crown sheets are usually made from $\frac{3}{4}$ to $\frac{7}{8}$ inch and the bolts about $1\frac{1}{4}$ inches in diameter. Sometimes, the stayrods *e* are set vertically.

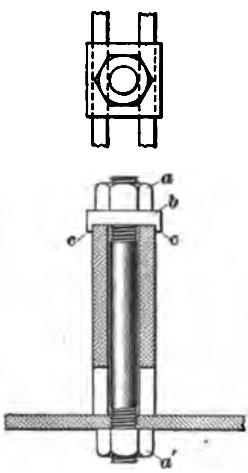


FIG. 35

39. Locomotive boilers are frequently made with the crown sheet and side firebox sheets in one piece, as shown in Fig. 36. In such a case, there is no ledge at *a* for supporting the crown bars. The crown sheet *b* is given a curved form as shown, and the crown bar *c* is made of a **T** iron, which is fastened to the crown sheet by means of bolts, with distance pieces placed between the crown bar and the plate. Vertical stays *d*, *d* suspended from the **T** iron *e* riveted to the top of the outer shell support the crown bar. In this construction, a staybolt is usually provided at the corner *a*, as there is a great tendency for the boiler shell to bulge outwards at the point *f* where the curvature changes.

A detail of the crown bar with its fastenings is shown in Fig. 37. The crown bar is shown in section at *a*; the button

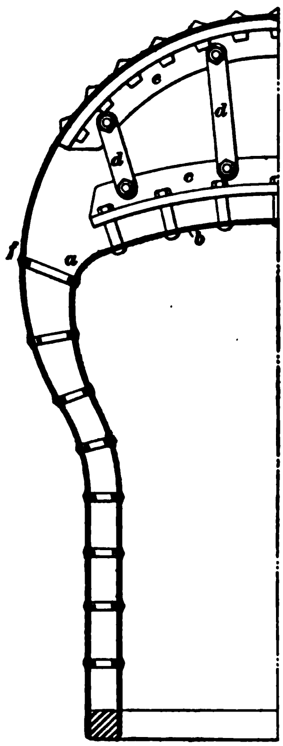


FIG. 36

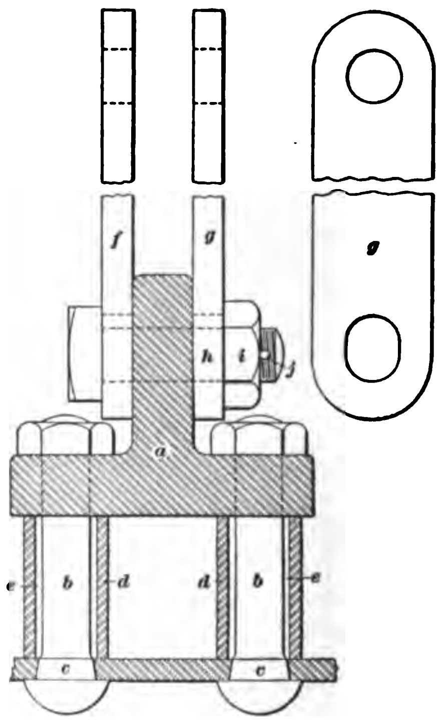


FIG. 37

head bolts *b*, *b* are tapered and made a driving fit at *c*, *c* and are fastened to the crown bar by means of nuts. The distance pieces, or spools, *d*, *d* are made of double extra-heavy wrought-iron pipe of such size as to provide a little space *e* around the bolt inside the spool. This arrangement permits free circulation between the crown bar and the crown sheet. The crown bar is supported from above at each point of support, by two stays, or straps, *f* and *g*, as shown. The straps are connected to the **T** irons by bolts *h* having a nut *i* and a cotter pin *j*. The stays *f* and *g* are provided with oblong holes at the lower ends, to allow for the more rapid cooling of the outer shell when the fire is low.

FLUE STAYS

40. Reinforcing Rings.—In marine boilers and some other internally fired boilers, the furnace is built in a flue that must resist the external pressure of the steam in the boiler. If the flue is straight throughout its entire length, its thickness must be such that it cannot rapidly transmit the heat from the fire to the water. It is therefore customary to strengthen flues at certain points of their length by using reinforcing rings, which are usually placed on the outside of the flues, so they will be away from the fire and in contact with the water and thus be prevented from burning out.

In Figs. 38 to 42 are shown sections of several good forms of furnace-flue stiffening devices. The form shown in Fig. 38 consists of an angle-iron ring *a* that encircles the flue and is

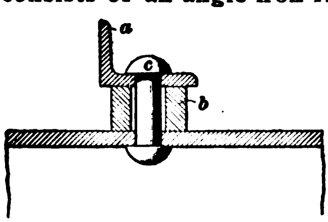


FIG. 38

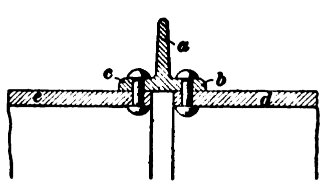


FIG. 39

separated from it by a spool *b* placed around each rivet *c*. The spools hold the ring away from the flue and thus provide for a free circulation of water between them. The circulation of water next to the flue protects it from injury by the fire. The construction shown in Fig. 39 consists of a welded **T**-iron ring *a*, with the legs *b* and *c* riveted to the sections *d* and *e* of the furnace flue. This form of construction is inexpensive, as the flue does not require flanging and the edges of the joint may be calked both inside and outside.

The flanged ring joint shown in Fig. 40 is made by flanging the ends *a* and *b* of the flue sections and riveting them together with a welded ring *c* placed between them. This method provides stiffness to resist collapsing pressure, and also makes a considerable allowance for expansion by means of the rounded bends in the plates at the flange. The rivet

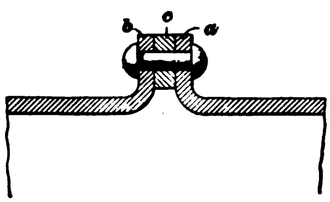


FIG. 40

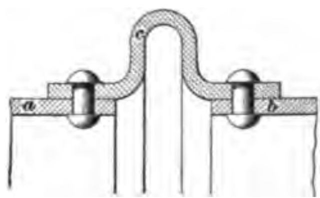


FIG. 41

heads are in the water and only one thickness of metal is directly exposed to the hot gases.

In Fig. 41 is shown a method of uniting the sections *a* and *b* of the flue by means of a **U**-shaped ring *c*. This ring provides for considerable longitudinal expansion or contraction, and besides stiffens the flue. The most common form of furnace flue, however, is shown in Fig. 42, in which the only stiffening is that due to corrugations in the metal of the flue itself.

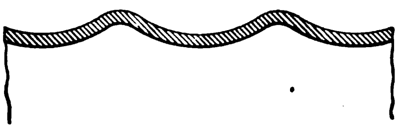


FIG. 42

41. Corrugated Flues. There are several types of **corrugated furnace flues,** and these differ only in the shape and arrangement of the corrugations. One form has corrugations with rounded top and bottom of equal radius; in another the corrugations are similar in form, but instead of each corrugation being independent and at right angles to the axis of the furnace flue, it forms a continuous spiral; in still another, the outwardly projecting corrugations are small and **V**-shaped, with a rounded top, while the inner projections are curved to a larger radius. This latter type is represented in Fig. 42, in which the corrugations are 8 inches between centers, and the variation of outside diameter of the flue is 3 inches.

42. The ends of corrugated flues are so shaped as to be easily riveted to the flanges of the boiler heads or combustion chambers. Figs. 43 and 44 show two of the most common types of ends used for making connections. In Fig. 43, the end *a* is designed to be riveted inside the flange on the boiler head, while the end *b* is riveted inside the flange on the combustion chamber. The end *a* of the flue shown in Fig. 44 is riveted inside the flange on the boiler head, but the other end is designed so that the part *b* is riveted to the side of the combustion chamber, while the part *c* is riveted to the bottom or end of the combustion chamber.

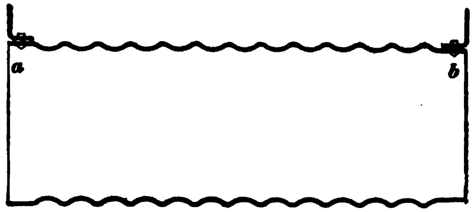


Fig. 43

The flues are considerably strengthened against collapsing under external pressure, and hence do not need to be so thick as straight flues. The corrugations also provide for longitudinal expansion and contraction. The flues can be of uniform thickness and will offer a greater heating surface; hence, they are more efficient in transmitting the heat to the water.

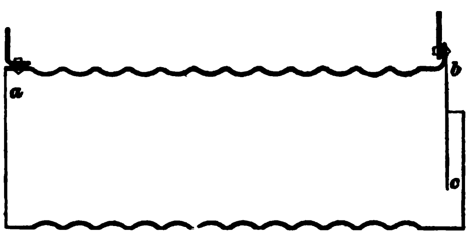


Fig. 44

The corrugations on the flues shown in Figs. 43 and 44 are made 8 inches between centers and with a depth of $1\frac{1}{2}$ inches, and the plain ends are not over 6 inches in length. This type is known as the **Morrison suspension corrugated flue.**

REINFORCING BOILER OPENINGS

43. Unless it must be circular, any opening in the cylindrical shell of a boiler should be longer in the girth than in the longitudinal direction. Boiler shells are weakened by the removal of material for openings therein, and the openings must therefore be reinforced. Reinforcing is done by riveting either one or two rings to the shell around the opening, in order to make up for the loss of strength occasioned by cutting out the metal for the opening. When only one reinforcing ring is used, it may be placed either inside or outside of the shell, but when two rings are used, one is placed next to the shell on the inside and the other next to the shell on the outside; both rings must not be on the same side.

In case piping is connected to the boiler at openings cut specially for that purpose, flanges or nozzles made with curves that fit the boiler outline are used and no reinforcing rings are required. Such connections are considered better than using reinforcing rings and threading them for pipe connections.

EXAMPLES OF STAYS IN BOILERS

44. Stays for Horizontal Return-Tubular Boiler. The details of the construction and arrangement of the stays for a 54-inch horizontal return-tubular boiler are shown in Fig. 45 (*a*) and (*b*). The method of staying the upper section of the tube-sheets consists in having two $6'' \times 2\frac{1}{2}''$ channel irons *a*, *a* riveted to the front and back heads *b* and *c* of the boiler, and five stayrods *d* $1\frac{5}{8}$ inches in diameter run from outside to outside of the heads, passing through the channel bars. The channels are secured to the heads by $\frac{13}{16}$-inch rivets. A $3'' \times 3''$ angle iron *e* is riveted to the top section of each head and to the upper channel *a*, transmitting its load to the channel. A cap nut *f* and a copper washer *g* are used on the outer ends of each stayrod to make a steam-tight joint, while a cast-iron washer *h* and a nut *i* are used on the stayrods to form the joints inside the heads. The

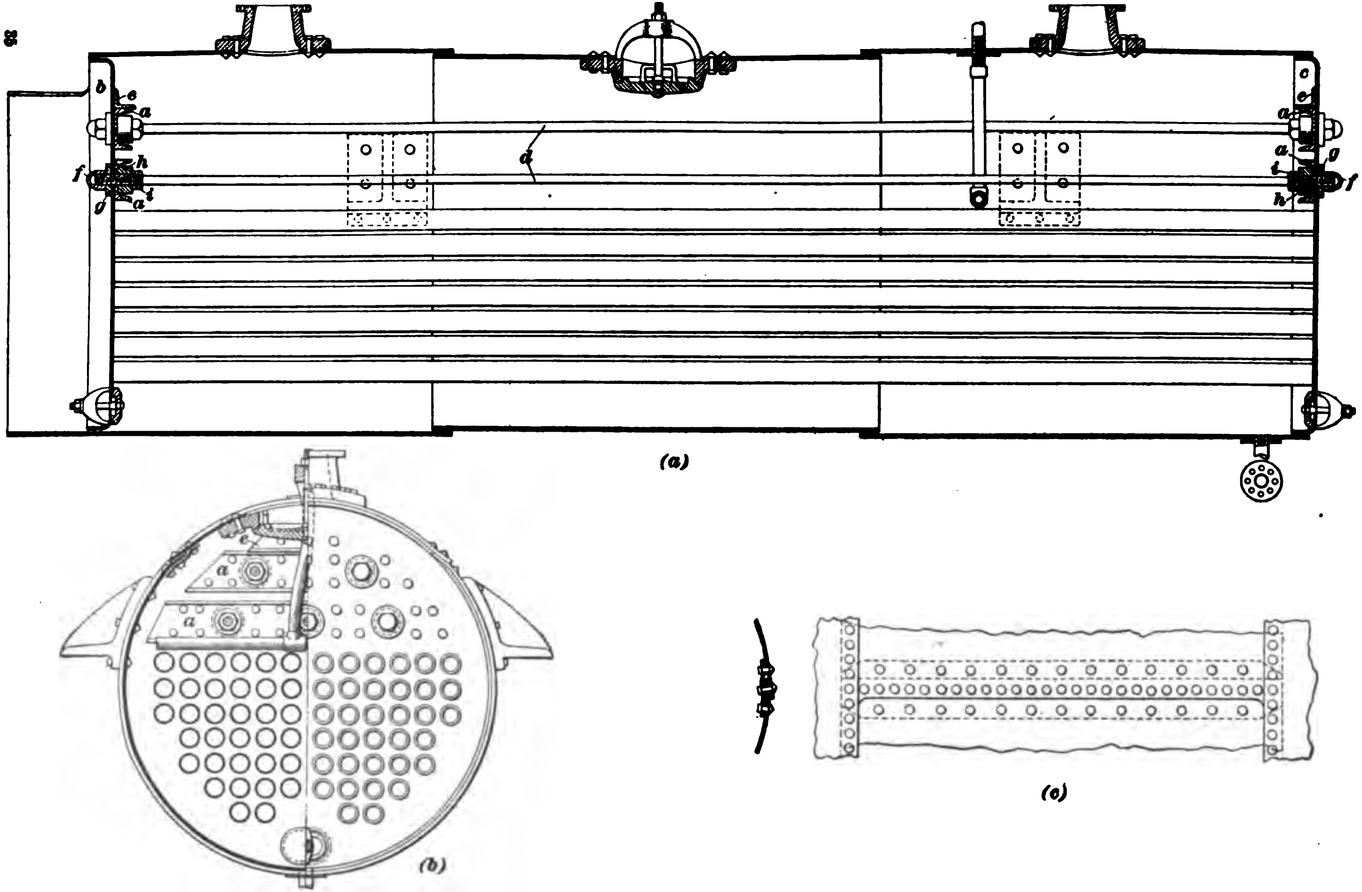


FIG. 45

ends of the stayrods where the threads are cut are upset $\frac{1}{4}$ inch larger than the body of the rod. This type of stay-rod is known as the **direct stay.**

By using channel irons and angle irons with a sufficient number of direct stays in the manner just described, the necessity of diagonal stays is avoided. The arrangement of the tubes is shown in Fig. 45 (*b*) and the riveting in (*c*). Both the tubes and the riveted joints act as stays to the boiler. The tubes are fitted to the tube-sheet, expanded to fill the hole, and beaded over on the outside; this is done to make the joint between the tube and tube-sheet steam-tight as well as to strengthen the head.

45. Stays for Marine Boiler.—Fig. 46 shows the details of construction and the method of staying a large double-ended marine boiler. The shell has a mean diameter of about 15 feet 3 inches, and is 16 feet 3 inches in length from outside to outside of the tube-sheets. There are eight furnaces of the corrugated type, with small corrugations on the exterior of the flue. The joint between the furnace flue *a* and the combustion chamber *b*, and also the joint between the furnace flue and the flange *c* on the boiler head, are made like those shown in Figs. 43 and 44, except that at *c* the flange is turned outwards. The flues are 3 feet 4 inches in diameter and 6 feet $10\frac{7}{8}$ inches long. The staying of this boiler is very simple and direct. The shell is made in three sections, or *strakes.* Each strake is composed of two sheets $1\frac{9}{32}$ inches thick. The heads are also made in three sections $\frac{7}{8}$ inch thick, and are so divided that all the furnaces come in one sheet, all the tubes in one, and the upper sections in one sheet. The furnace-end sheets are stayed by the corrugated flues, which are flanged out and riveted to the sides of the combustion chamber, as shown at *d*, and riveted to flanges in the end plates, as shown at *c*. The tube-sheets are stayed by the tubes, which are $2\frac{1}{2}$ inches outside diameter. There are 792 common expanded tubes of a thickness equal to No. 8 Birmingham wire gauge, and 280 extra-heavy screwed stay tubes, which are $\frac{3}{16}$ inch thick

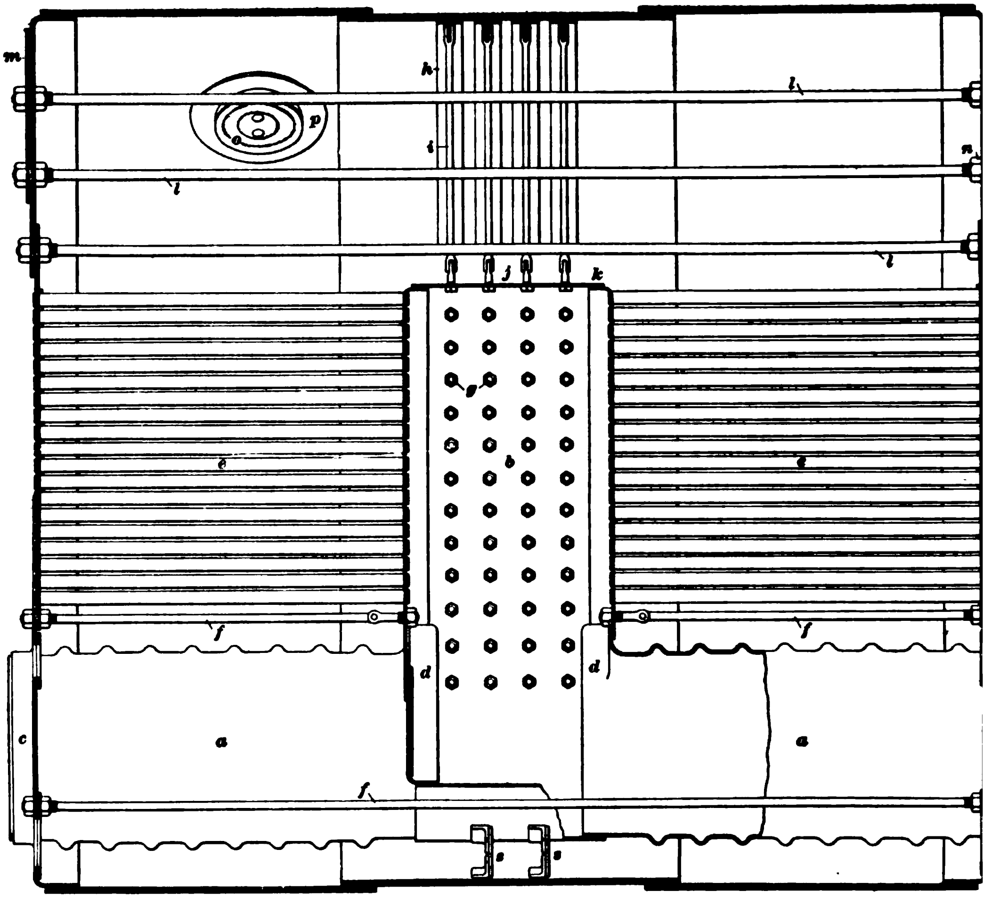
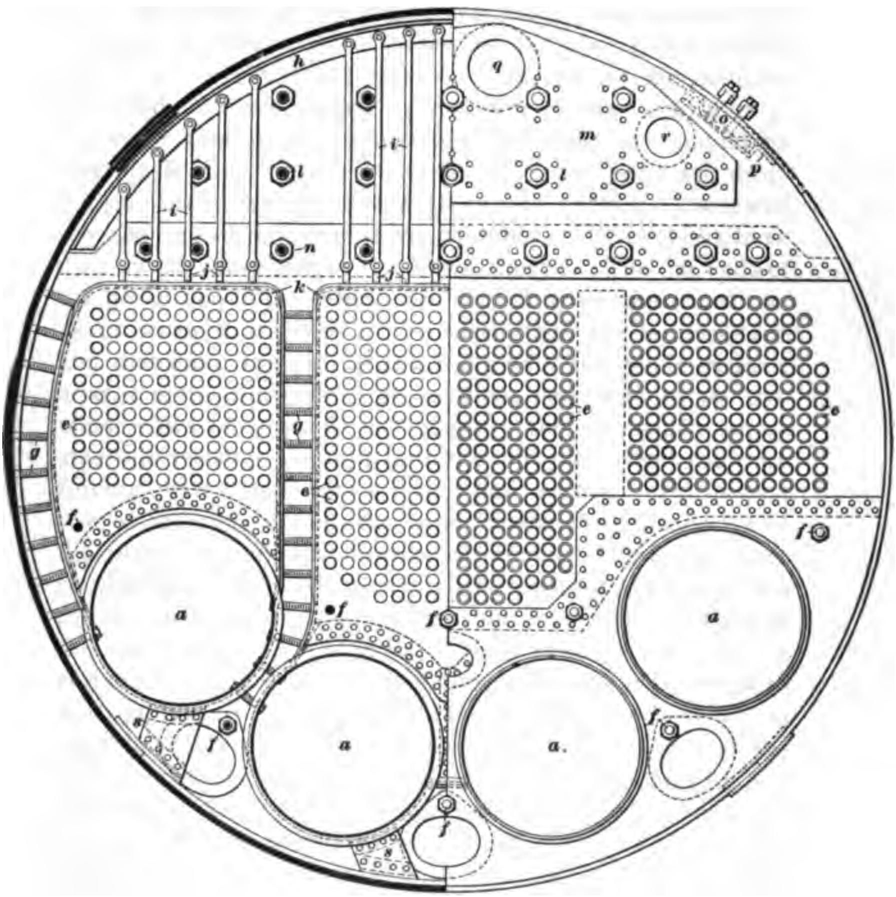


FIG.


h
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and have upset, threaded ends screwed directly into the tube-sheet. This method can be used only when the tube-sheets are thick; with thin tube-sheets and combustion chambers, a flat nut is used on the end of the tube. The sides of the combustion chambers are stayed by the tubes *e*, the furnace flues *a*, and eight direct stays *f*. The ends of the combustion chambers are stayed by common screwed staybolts *g*, while the tops of the combustion chambers are stayed by riveting **T** irons *h* to the inside of the boiler shell and dropping $1\frac{1}{8}$-inch diameter swing stays *i* to eyebolts *j* screwed into the tops of the combustion chamber. The eyebolts have nuts on the under side of the crown sheet *k*. The flat ends of the boiler above the tubes *e* are stayed by twenty-one direct upset screwed stayrods *l*. To strengthen the boiler heads and distribute the pull of the bolts over the whole surface, a large plate *m* covering the segment embraced by the two upper rows of stayrods is riveted to the end sheets, these stayrods passing through both the end plates and the reinforcing plates, with nuts *n* inside and outside. The lowest row of stayrods pierces the boiler heads through the lap of the plates forming the horizontal joints. There are three combustion chambers *b*; two furnaces from each end lead into the middle one and one furnace from each end into each of the outer ones. The object of this construction is to prevent chilling the whole boiler when the fire-doors are opened to any one furnace. Four $10'' \times 14''$ manholes are provided in each head, being reinforced by plates and stayed by the direct stays *f*. A manhole *o* reinforced by a flanged ring *p* is provided in the top of the shell. Flanged openings *q* and *r* through the top segments are provided for the steam connections. The combustion chambers and rear ends of the furnaces are supported on gusset stays *s*. The joints in the shell in the different strakes are placed 90° from each other and are triple-riveted butt joints with double cover-plates.

46. Stays for Locomotive Boiler.—In Figs. 47 and 48, the method of staying a locomotive boiler is illustrated. The front and rear heads above and at the sides of the

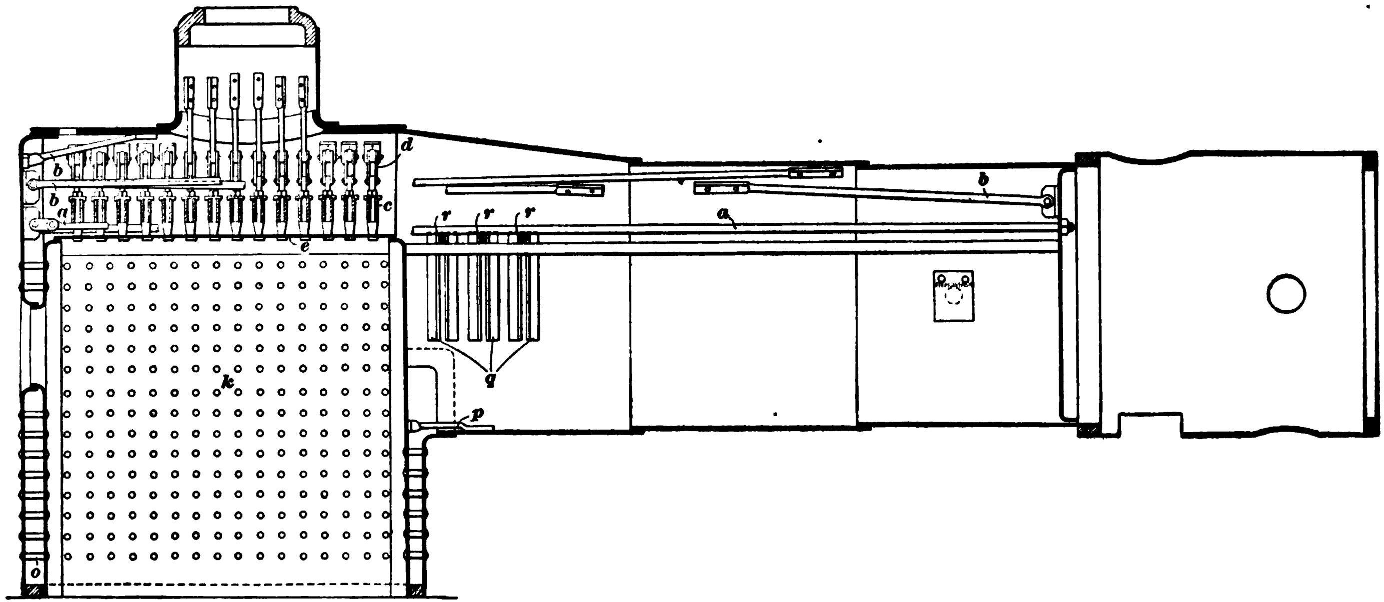


FIG. 47

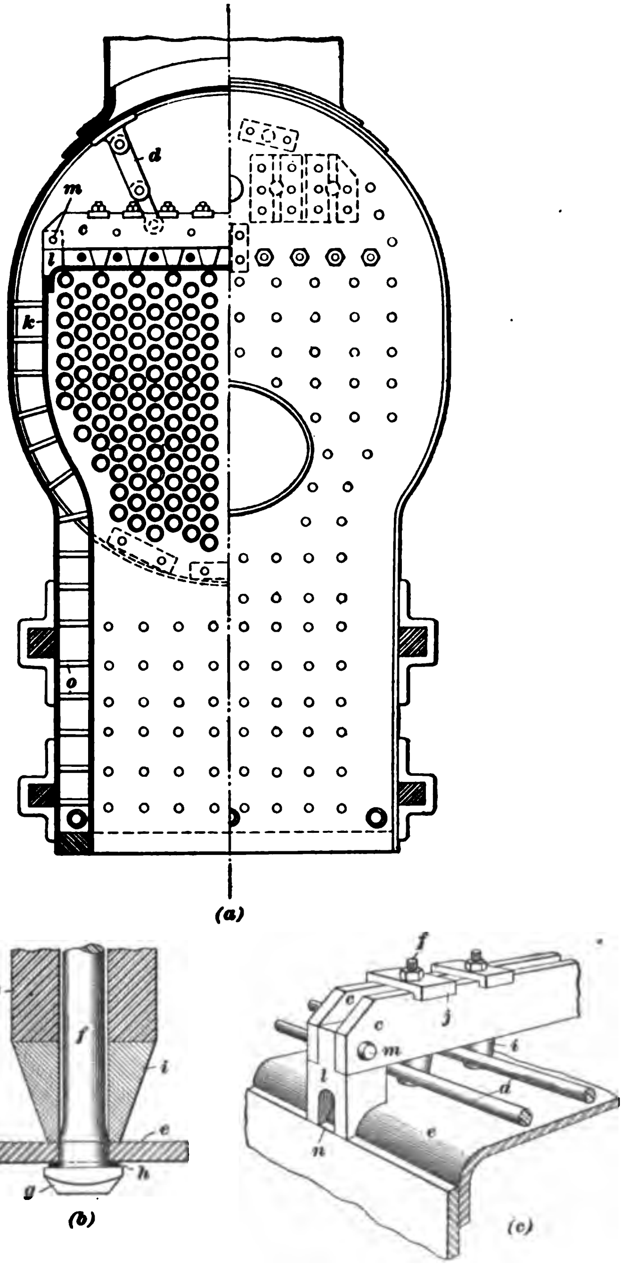


 Fig. 48

fire-tubes, Fig. 47, are stayed by direct stays *a* and diagonal braces *b* riveted to the shell and having pin connections to angle irons riveted to the head. The flat top of the crown sheet is stayed by means of girders *c* and swinging braces *d*.

An enlarged detail of the method of securing the crown sheet *e* to the crown bars *c* is shown in Fig. 48 (*b*), and consists of the $\frac{7}{8}$-inch stay bolts *f* passing between the crown bars and having a flat square head *g* in the combustion chamber. A copper washer *h* is used under the bolt head, and a conical spool *i* between the lower sides of the crown bars and top of the crown sheet resists the upward pull of the bolts. These spools are made conical, with the small end of the frustum against the crown sheet, to allow the water to circulate around the bolt as closely as possible and avoid burning the sheet because of extra thickness of metal at these places.

The crown bars are prevented from spreading by lips *j* on the top washer, as shown in Fig. 48 (*c*), which is an enlarged detail showing the method of supporting the ends of the crown bars on the side sheets *k* of the combustion chamber. The support consists of a wrought-iron chair *l*, into which the ends of the crown bars *c* are mortised and secured by a single rivet *m*. The space *n* is cut out of the bottom of the chair *l* to allow a circulation of water under it and thus prevent the crown sheet from burning. Common screw staybolts *o*, Fig. 48 (*a*), are used all around the combustion chamber, to stay the inner walls to the outer shell of the boiler.

The front end of the combustion chamber unsupported by the fire-tubes has diagonal stays, as shown at *p*, Fig. 47. The flat sides of the boiler shell, where they are not supported by staybolts connected to the combustion chamber, are stayed by riveting angle irons *q* to the shell at opposite sides of the boiler and running cross-stays *r* over the tops of the tubes, thus tying the angles together.

BOILER SUPPORTS

47. Horizontal return-tubular boilers are usually supported on the brick side walls of the boiler settings by lugs or brackets similar to the one shown in Fig. 49. Four of

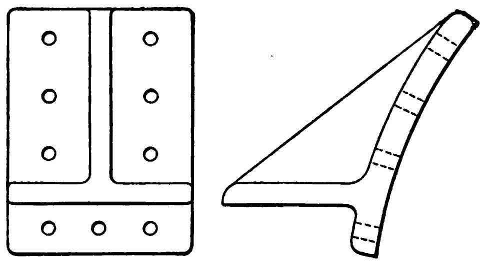


FIG. 49

these brackets, two on each side, are placed on boilers up to 16 feet in length, and six on boilers 18 or 20 feet in length. The lugs are riveted to the boiler shell, slightly above the middle. The two near the front end are placed on iron plates on the wall of the setting when the boilers are installed, and the others rest on iron rollers on iron plates. The purpose of the rollers is to permit the boiler to expand without setting up unnecessary stresses at the lugs or rivets. The arrangement of the rivets in the lug should be such as to give greatest strength and rigidity; the method shown in Fig. 49 is one of the best.

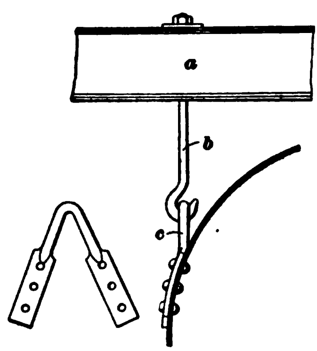


FIG. 50

Boilers are sometimes supported by being suspended from pairs of channel-iron beams resting on the side walls of the setting. A detail of such a support is shown in Fig. 50. The beams *a* rest on the side walls, the hooks *b* are supported from the beams by washers and nuts, and the loops or straps *c* are riveted to the boiler shell. The straps can be placed high enough to clear the boiler setting, and yet not much above, so that they will be as nearly vertical as possible. The only stresses that will then need to be considered in connection with the straps are the shearing stress of the rivets and the tension of the strap.

Sometimes, a **U** bolt with two nuts bearing on the cross-beams is used instead of the hook *b*. With this device there is no tendency to open when the load is applied, as in the case with the hook.

STEAM-BOILER DESIGN
(PART 2)

BOILER CALCULATIONS

STRENGTH OF BOILER SHELLS AND FLUES

STRENGTH UNDER INTERNAL PRESSURE

1. General Formula.—The strength of a boiler or of a flue subjected to internal fluid pressure may, according to the principles of strength of materials, be computed by the use of the formula $pD = 2tS_t$. In this formula,

p = pressure, in pounds per square inch;
D = diameter of boiler or flue, in inches;
t = thickness of shell, in inches;
S_t = ultimate tensile strength of the material, in pounds per square inch.

Introducing a factor of safety f in the above formula and solving for the pressure,

$$p = \frac{2tS_t}{Df}$$

2. The factor of safety of a boiler is the ratio of the stress at which rupture would occur to the stress under ordinary working conditions. In steam-boiler work, it is commonly taken at from 4 to 6, though some city ordinances fix the factor of safety as 5 for all parts of a boiler. When some parts are so peculiarly shaped that their strengths are difficult to calculate, a comparatively large factor of safety should be used. Also, suitable allowance should be made for wasting away by corrosion, which is much more rapid in some parts of a boiler than in others. Thus, staybolts and stayrods are more susceptible to corrosion than is the boiler shell. The joints of the shell, being protected by the rivets and cover-plates, are less affected by corrosion than is the remainder of the shell. Stays are usually made $\frac{1}{8}$ inch larger than the calculated diameter, to allow for the loss due to corrosion.

3. Thickness of Shell.—When designing boilers, the tensile strength of the material is generally indicated in specifications, and the factor of safety, diameter, and pressure per square inch are decided on, according to the work the boiler has to perform. The first unknown quantity to be determined is the thickness t of the shell. Solving the formula in Art. **1** for t,

$$t = \frac{f p D}{2 S_t} \qquad (1)$$

All plates used for making boiler shells should be tested, and should have their ultimate tensile strengths stamped on them. The lowest value stamped on the plate used should be substituted for S_t in formula **1.**

The riveted joints of a boiler shell commonly have from 50 to 90 per cent. of the strength of the solid plate, although, in special kinds of joints, the percentage may be as great as 95. This percentage is called the *efficiency of the joint*, and may be denoted by y. It is evident that, to obtain the true strength of the shell, the strength of the weakest part, that is, the riveted joint, should be calculated, or, what is the same thing, the quantity S_t in formula **1** should be multiplied by the efficiency y. Formula **1** then becomes

$$t = \frac{f p D}{2 S_t y} \qquad (2)$$

Should it be required to find the pressure per square inch that a given boiler will carry, formulas **1** and **2** should be solved for p. Then,

$$p = \frac{2 t S_t}{f D} \qquad (3)$$

$$p = \frac{2 t S_t y}{f D} \qquad (4)$$

EXAMPLE.—Find the safe steam pressure that may be carried by a boiler 44 inches in diameter and $\frac{5}{16}$ inch in thickness, the efficiency of the joint being 64 per cent., the ultimate tensile strength 55,000 pounds per square inch, and the factor of safety 5.

SOLUTION.—By formula **4**,

$$p = \frac{2\,t\,S_t\,y}{f\,D} = \frac{2 \times \frac{5}{16} \times 55{,}000 \times .64}{5 \times 44} = 100 \text{ lb. per sq. in.} \quad \text{Ans.}$$

EXAMPLE.—Required, the thickness of a boiler shell that is 5 feet in diameter, under a pressure of 90 pounds per square inch. The shell is of steel having a tensile strength of 60,000 pounds, the factor of safety is 4, and the efficiency of the joint is assumed to be 72 per cent.

SOLUTION.—By formula **2**,

$$t = \frac{f\,p\,D}{2\,S_t\,y} = \frac{4 \times 90 \times 5 \times 12}{2 \times 60{,}000 \times .72} = \tfrac{1}{4} \text{ in.} \quad \text{Ans.}$$

EXAMPLES FOR PRACTICE

1. Assuming a factor of safety of 6, and an efficiency of joint of 55 per cent., find the pressure that may be carried by an iron boiler, 3 feet 6 inches in diameter and $\frac{1}{4}$ inch thick, the tensile strength of the plates being 50,000 pounds per square inch.
Ans. 54.6 lb. per sq. in., nearly

2. Calculate the necessary thickness of a shell 32 inches in diameter, the safe working stress being 10,000 pounds per square inch, and the efficiency of the joint 60 per cent. The working pressure is 120 pounds per square inch. Ans. $\frac{5}{16}$ in.

NOTE.—When the calculated thickness comes out as a decimal, take the nearest sixteenth. For example, if the calculated thickness is .32 inch, take $\frac{5}{16}$ inch as the proper dimension.

3. Find the thickness of a shell 54 inches in diameter, if the steel plates have a tensile strength of 63,000 pounds per square inch. The efficiency of the joint is 70 per cent., and 6 is to be taken as the factor of safety. Steam pressure is 150 pounds per square inch. Ans. $\frac{9}{16}$ in.

4. Compute the thickness of the shell of a marine boiler $13\frac{1}{2}$ feet in diameter under a steam pressure of 160 pounds per square inch. The safe working stress of the steel is 12,000 pounds per square inch, and the efficiency of the seam is 80 per cent. Ans. $1\frac{3}{8}$ in.

5. What pressure may be safely carried by a welded water tube, 4 inches in diameter and $\frac{5}{32}$ inch thick? The safe working stress is 11,000 pounds per square inch. Ans. $859\frac{3}{8}$ lb. per sq. in.

STRENGTH UNDER EXTERNAL PRESSURE

4. Strength of Flues and Tubes.—When a cylinder is subjected to internal pressure, as a boiler shell, the pressure, being equal in all directions, tends to maintain the vessel in a truly cylindrical shape. When, however, the vessel is subjected to *external* pressure, as in the case of flues, the pressure tends to distort the vessel from its true shape. Thus, supposing a cylinder to be slightly elliptical, as shown at *a*, Fig. 1, an internal pressure would tend to return it to its

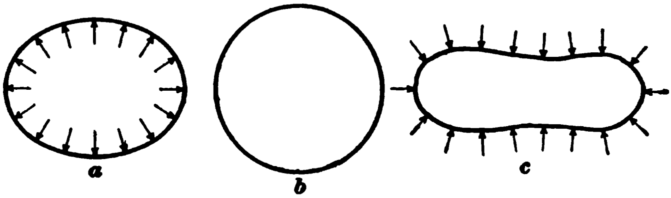


FIG. 1

original circular form, as shown at *b*, while an external pressure would tend to make it collapse, as shown at *c*.

The formulas for the strength and thickness of flues to resist external pressure cannot be deduced by abstract reasoning as they were for internal pressures, and such formulas must therefore be based on experiment.

Numerous experiments were made by Prof. R. T. Stewart to determine the collapsing pressures of lap-welded steel tubes from 3 to 10 inches in diameter. His observations showed that the length of the tube has practically no effect on the collapsing pressure, provided the length is not less than six times the diameter of the tube. The formulas for the collapsing pressures of modern lap-welded Bessemer-steel tubes, as based on his experiments, are as follows:

$$p = 86{,}670\,\frac{t}{d} - 1{,}386 \qquad (1)$$

and

$$p = 1{,}000\left(1 - \sqrt{1 - 1{,}600\,\frac{t^2}{d^2}}\right), \qquad (2)$$

in which p = collapsing pressure, in pounds per square inch;

t = thickness of tube, in inches;

d = outside diameter of tube, in inches.

Formula **1** is to be used when the value of p is greater than 581 pounds or when the ratio $\frac{t}{d}$ is greater than .023, while formula **2** is to be used when the values of p and $\frac{t}{d}$ are less than 581 and .023, respectively.

It may at times be desired to use these formulas to find the thickness or the diameter of a tube. When the pressure is greater than 581 pounds, or $\frac{t}{d}$ is greater than .023, the thickness is found by transposing formula **1**, thus:

$$t = \frac{d\,(p + 1{,}386)}{86{,}670} \qquad (3)$$

When the pressure is less than 581 pounds, or $\frac{t}{d}$ is less than .023, the thickness is found by transposing formula **2**, thus:

$$t = \frac{d}{40}\sqrt{1 - \left(1 - \frac{p}{1{,}000}\right)^2} \qquad (4)$$

When the pressure is greater than 581 pounds, or $\frac{t}{d}$ is greater than .023, the diameter is found by transposing formula **1**, thus:

$$d = \frac{86{,}670\,t}{p + 1{,}386} \qquad (5)$$

When the pressure is less than 581 pounds, or $\frac{t}{d}$ is less than .023, the diameter is found by transposing formula **2**, thus:

$$d = \frac{40\,t}{\sqrt{1 - \left(1 - \frac{p}{1{,}000}\right)^2}} \qquad (6)$$

EXAMPLE 1.—What is the collapsing pressure of a 3-inch lap-welded steel boiler tube .109 inch in thickness?

SOLUTION.—Inasmuch as the pressure is unknown, it becomes necessary to find the ratio of $\frac{t}{d}$ before deciding which formula to use. As $\frac{t}{d} = \frac{.109}{3} = .0363$, which is greater than .023, use formula **1**. Then,

$p = 86{,}670 \times \frac{t}{d} - 1{,}386 = 86{,}670 \times \frac{.109}{3} - 1{,}386 = 1{,}763$ lb. per sq. in. Ans.

EXAMPLE 2.—Find the thickness of a lap-welded steel tube 10 inches in diameter that will collapse at a pressure of 580 pounds per square inch.

SOLUTION.—Applying formula 4,

$$t = \frac{d}{40}\sqrt{1 - \left(1 - \frac{p}{1,000}\right)^2} = \frac{10}{40}\sqrt{1 - \left(1 - \frac{580}{1,000}\right)^2} = .2269 \text{ inch. Ans.}$$

EXAMPLE 3.—Find the diameter of a tube .14 inch thick that will collapse at a pressure of 1,000 pounds per square inch.

SOLUTION.—Applying formula 5,

$$d = \frac{86,670\,t}{p + 1,386} = \frac{86,670 \times .14}{1,000 + 1,386} = 5.085, \text{ say 5, in. Ans.}$$

EXAMPLES FOR PRACTICE

1. Find the collapsing pressure of a $3\frac{1}{2}$-inch lap-welded boiler tube .12 inch thick. Ans. 1,586 lb. per sq. in.

2. Find the thickness of a 4-inch tube that will just collapse under a pressure of 1,000 pounds per square inch. Ans. .11 in.

3. Find the collapsing pressure of a 9-inch lap-welded boiler tube .18 inch thick. Ans. 400 lb. per sq. in.

PROPORTIONS AND STRENGTH OF RIVETED JOINTS

PROPORTIONS OF RIVETED JOINTS

5. Tables I and II give the proportions of riveted joints used by some of the leading boilermakers in the United States.

Table I is applicable to lap joints and butt joints with single cover-plate. For boiler plates more than $\frac{1}{2}$ inch thick, butt joints with two cover-plates are recommended.

The efficiencies in these tables are obtained from the formula $y = \frac{(h - d)\,t\,S_t}{h\,t\,S_t} = \frac{h - d}{h}$.

6. The Hartford Steam Boiler Inspection and Insurance Company has designed and recommends the joints given in Table II, the designs being for steel plate and steel rivets. With the values of the ultimate tensile and shearing strengths given, the efficiencies of these joints are found in the last column.

TABLE I

PITCH AND EFFICIENCY OF RIVETED JOINTS

Tensile Strength of Plate (Steel), 55,000 Pounds; Rivets (Steel), 45,000 Pounds

Thickness of Plate, in Inches t	Diameter of Rivet, in Inches d'	Diameter of Hole, in Inches d	Pitch, in Inches h		Efficiency of Joint y	
			Single	Double	Single	Double
$\frac{1}{4}$	$\frac{5}{8}$	$\frac{11}{16}$	2	3	.66	.77
$\frac{5}{16}$	$\frac{11}{16}$	$\frac{3}{4}$	$2\frac{1}{16}$	$3\frac{1}{8}$	.64	.76
$\frac{3}{8}$	$\frac{3}{4}$	$\frac{13}{16}$	$2\frac{1}{8}$	$3\frac{1}{4}$	.62	.75
$\frac{7}{16}$	$\frac{13}{16}$	$\frac{7}{8}$	$2\frac{3}{16}$	$3\frac{3}{8}$	.60	.74
$\frac{1}{2}$	$\frac{7}{8}$	$\frac{15}{16}$	$2\frac{1}{4}$	$3\frac{1}{2}$	.58	.73

STRENGTH OF RIVETED JOINTS

7. Table III gives the formulas commonly employed to determine the strength of riveted joints, as well as formulas for the diameter and pitch of the rivets. In these formulas,

d = diameter of rivet, in inches, after riveting;

h = pitch of rivets in outer row, in inches;

t = thickness of plate, in inches;

S_t = ultimate tensile strength, in pounds per square inch of plate (50,000 for wrought iron, and 55,000 for steel);

S_s = ultimate shearing strength per square inch of rivets (40,000 pounds for wrought iron, and 45,000 for steel);

S_c = ultimate crushing strength per square inch of plate (80,000 pounds for wrought iron, and 90,000 pounds for steel);

R_t = ultimate strength of the riveted joint for a width equal to h, in tension;

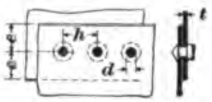
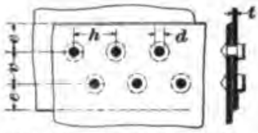
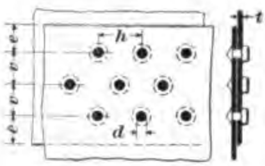
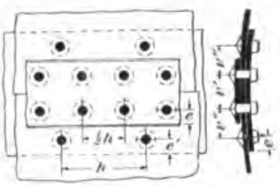
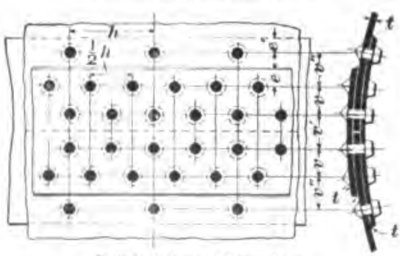
R_s = ultimate strength of the riveted joint for a width equal to h, in shear;

R_c = ultimate strength of the riveted joint for a width equal to h, in compression;

TABLE II

PROPORTIONS OF RIVETED JOINTS

Tensile Strength of Plates and Straps (Steel) = 55,000 Pounds; Rivets (Steel) 42,000 Pounds Single Shear, 77,700 Pounds Double Shear

Type of Joint	t	d'	d	h	$\frac{1}{2}h$	e	e'	v	v'	v''	t'	t''	Efficiency Per Cent.
 Single-Riveted Lap Joint	$\frac{1}{4}$	$\frac{9}{16}$	$\frac{5}{8}$	$1\frac{1}{2}$		$\frac{15}{16}$							58.3
	$\frac{5}{16}$	$\frac{11}{16}$	$\frac{3}{4}$	$1\frac{13}{16}$		$1\frac{1}{8}$							58.6
	$\frac{3}{8}$	$\frac{13}{16}$	$\frac{7}{8}$	$2\frac{1}{8}$		$1\frac{5}{16}$							*57.6
	$\frac{7}{16}$	$\frac{15}{16}$	1	$2\frac{3}{8}$		$1\frac{1}{2}$							*57.7
	$\frac{1}{2}$	1	$1\frac{1}{16}$	$2\frac{15}{32}$		$1\frac{19}{32}$							*54.8
 Double-Riveted Lap Joint	$\frac{1}{4}$	$\frac{9}{16}$	$\frac{5}{8}$	$2\frac{7}{16}$		$\frac{15}{16}$		$1\frac{1}{2}$					74.3
	$\frac{5}{16}$	$\frac{5}{8}$	$\frac{11}{16}$	$2\frac{1}{2}$		$1\frac{1}{32}$		$1\frac{9}{16}$					72.5
	$\frac{3}{8}$	$\frac{3}{4}$	$\frac{13}{16}$	$2\frac{7}{8}$		$1\frac{7}{32}$		$1\frac{3}{4}$					71.7
	$\frac{7}{16}$	$\frac{7}{8}$	$\frac{15}{16}$	$3\frac{1}{4}$		$1\frac{13}{32}$		2					71.1
	$\frac{1}{2}$	$\frac{15}{16}$	1	$3\frac{3}{8}$		$1\frac{1}{2}$		$2\frac{1}{16}$					70.3
 Triple-Riveted Lap Joint	$\frac{1}{4}$	$\frac{9}{16}$	$\frac{5}{8}$	$2\frac{7}{8}$		$\frac{15}{16}$		$1\frac{3}{4}$					78.2
	$\frac{5}{16}$	$\frac{5}{8}$	$\frac{11}{16}$	3		$1\frac{1}{16}$		$1\frac{7}{8}$					77.0
	$\frac{3}{8}$	$\frac{3}{4}$	$\frac{13}{16}$	$3\frac{1}{4}$		$1\frac{7}{32}$		$2\frac{1}{8}$					75.0
	$\frac{7}{16}$	$\frac{13}{16}$	$\frac{7}{8}$	$3\frac{1}{2}$		$1\frac{5}{16}$		$2\frac{1}{4}$					75.0
	$\frac{1}{2}$	$\frac{7}{8}$	$\frac{15}{16}$	$3\frac{3}{4}$		$1\frac{13}{32}$		$2\frac{3}{8}$					75.0
 Double-Riveted Butt Joint	$\frac{1}{4}$	$\frac{5}{8}$	$\frac{11}{16}$	4	2	$1\frac{1}{32}$	$1\frac{1}{32}$		$2\frac{1}{16}$	$2\frac{1}{16}$	$\frac{7}{32}$	$\frac{7}{32}$	82.8
	$\frac{5}{16}$	$\frac{11}{16}$	$\frac{3}{4}$	$4\frac{1}{2}$	$2\frac{1}{4}$	$1\frac{1}{8}$	$1\frac{1}{8}$		$2\frac{1}{4}$	$2\frac{1}{4}$	$\frac{1}{4}$	$\frac{1}{4}$	83.3
	$\frac{3}{8}$	$\frac{3}{4}$	$\frac{13}{16}$	$4\frac{25}{32}$	$2\frac{25}{64}$	$1\frac{7}{32}$	$1\frac{7}{32}$		$2\frac{7}{16}$	$2\frac{7}{16}$	$\frac{5}{16}$	$\frac{5}{16}$	83.0
	$\frac{7}{16}$	$\frac{13}{16}$	$\frac{7}{8}$	5	$2\frac{1}{2}$	$1\frac{5}{16}$	$1\frac{5}{16}$		$2\frac{5}{8}$	$2\frac{5}{8}$	$\frac{3}{8}$	$\frac{3}{8}$	82.5
	$\frac{1}{2}$	$\frac{7}{8}$	$\frac{15}{16}$	$5\frac{1}{8}$	$2\frac{9}{16}$	$1\frac{13}{32}$	$1\frac{13}{32}$		$2\frac{13}{16}$	$2\frac{13}{16}$	$\frac{7}{16}$	$\frac{7}{16}$	81.7
 Triple-Riveted Butt Joint	$\frac{1}{4}$	$\frac{1}{2}$	$\frac{9}{16}$	$4\frac{1}{2}$	$2\frac{1}{4}$	$\frac{27}{32}$	$\frac{27}{32}$	$1\frac{3}{8}$	$1\frac{11}{16}$	$1\frac{11}{16}$	$\frac{3}{16}$	$\frac{3}{16}$	87.5
	$\frac{5}{16}$	$\frac{11}{16}$	$\frac{3}{4}$	$6\frac{1}{4}$	$3\frac{1}{8}$	$1\frac{1}{8}$	$1\frac{1}{8}$	$1\frac{7}{8}$	$2\frac{1}{4}$	$2\frac{1}{4}$	$\frac{1}{4}$	$\frac{1}{4}$	88.0
	$\frac{3}{8}$	$\frac{3}{4}$	$\frac{13}{16}$	$6\frac{1}{2}$	$3\frac{1}{4}$	$1\frac{7}{32}$	$1\frac{7}{32}$	2	$2\frac{7}{16}$	$2\frac{7}{16}$	$\frac{5}{16}$	$\frac{5}{16}$	87.5
	$\frac{7}{16}$	$\frac{7}{8}$	$\frac{15}{16}$	$6\frac{3}{4}$	$3\frac{3}{8}$	$1\frac{13}{32}$	$1\frac{13}{32}$	$2\frac{1}{32}$	$2\frac{13}{16}$	$2\frac{13}{16}$	$\frac{3}{8}$	$\frac{3}{8}$	86.1
	$\frac{1}{2}$	1	$1\frac{1}{16}$	$7\frac{1}{2}$	$3\frac{3}{4}$	$1\frac{19}{32}$	$1\frac{19}{32}$	$2\frac{1}{4}$	$3\frac{3}{16}$	$3\frac{3}{16}$	$\frac{3}{8}$	$\frac{3}{8}$	85.8
	$\frac{9}{16}$	1	$1\frac{1}{16}$	$7\frac{9}{16}$	$3\frac{25}{32}$	$1\frac{19}{32}$	$1\frac{19}{32}$	$2\frac{9}{32}$	$3\frac{3}{16}$	$3\frac{3}{16}$	$\frac{7}{16}$	$\frac{7}{16}$	85.9
	$\frac{5}{8}$	1	$1\frac{1}{16}$	$7\frac{3}{4}$	$3\frac{7}{8}$	$1\frac{19}{32}$	$1\frac{19}{32}$	$2\frac{15}{16}$	$3\frac{3}{16}$	$3\frac{3}{16}$	$\frac{15}{32}$	$\frac{15}{32}$	86.3

*Least efficiency lies in rivet instead of plate.

R_{st} = ultimate strength of the riveted joint for a width equal to h, in shear and in tension, as by shearing off the outer row of rivets and tearing the plate along the adjacent row;

R_{cs} = ultimate strength of the riveted joint for a width equal to h, in compression and shear, as by shearing off the outer row of rivets and crushing out along the adjacent row.

It should be clearly understood that the values of S_t, S_s, and S_c just given are simply average values. In case the actual strengths of the materials are determined by tests, such known values should be used. Or, if a boiler is to be designed according to a given set of specifications, the values of S_t, S_s, and S_c given therein must be used.

The distance from the center of the rivet to the nearest edge of the plate should be at least $1\frac{1}{2}d$. For single-riveted joints in longitudinal seams recent investigators recommend that this distance be made about $2d$.

The distance between adjacent rows of rivets, called the *transverse pitch*, should not be less than $2d$, and is preferably made $2.5d$. This distance should in all cases be great enough so that the rivet die, in forming the head of one rivet, will not strike the head of the adjoining rivet.

8. To show the application of the formulas in Table III and those that precede the table, let it be required to design and calculate the strength of the joints of a boiler 4 feet in diameter, to carry a pressure of 120 pounds per square inch, the longitudinal seam being a double-riveted butt joint with unequal cover-plates and the girth seam a single-riveted lap joint. The steel for the shell has a tensile strength S_t of 53,000 pounds, and a crushing strength S_c of 90,000 pounds. The shearing strength S_s of the steel rivets is 44,000 pounds, and the factor of safety is to be 6 for all material.

The thickness of the shell, according to formula **1**, Art. **3**,

$$t = \frac{fpd}{2S_t} = \frac{6 \times 120 \times 48}{2 \times 53,000} = .326 \text{ inch}$$

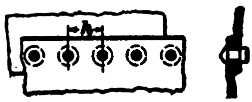
(*Continued on page 13*)

TABLE III

STRENGTHS OF RIVETED JOINTS

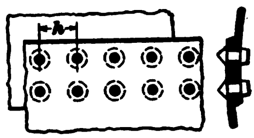
TYPE OF JOINT AND FORMULAS FOR STRENGTH

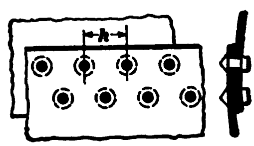
Single-riveted lap joint.



$$R_t = (h-d)\,t\,S_t$$
$$R_s = .7854\,d^2\,S_s$$
$$R_c = d\,t\,S_c$$
$$d = 1.27\,\frac{t\,S_c}{S_s}$$
$$h = \frac{.7854\,d^2\,S_s}{t\,S_t} + d$$

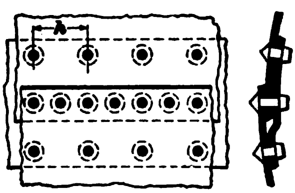
Double-riveted lap joint. Chain or staggered riveting.





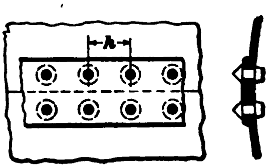
$$R_t = (h-d)\,t\,S_t$$
$$R_s = 1.57\,d^2\,S_s$$
$$R_c = 2\,d\,t\,S_c$$
$$d = 1.27\,\frac{t\,S_c}{S_s}$$
$$h = \frac{1.57\,d^2 S_s}{t\,S_t} + d$$

Single-riveted lap joint, with cover-plate.



$$R_t = (h-d)\,t\,S_t$$
$$R_s = 2.36\,d^2\,S_s$$
$$R_c = 3\,d\,t\,S_c$$
$$R_{st} = (h-2\,d)\,t\,S_t + .7854\,d^2\,S_s$$
$$R_{cs} = 2\,d\,t\,S_c + .7854\,d^2\,S_s$$
$$d = 1.27\,\frac{t\,S_t}{S_s}$$
$$h = \frac{2.36\,d^2\,S_s}{t\,S_t} + d$$

Single-riveted butt joint, with one cover-plate.

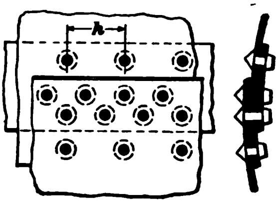


$$R_t = (h-d)\,t\,S_t$$
$$R_s = .7854\,d^2\,S_s$$
$$R_c = d\,t\,S_c$$
$$d = 1.27\,\frac{t\,S_c}{S_s}$$
$$h = \frac{.7854\,d^2\,S_s}{t\,S_t} + d$$

TABLE III—(*Continued*)

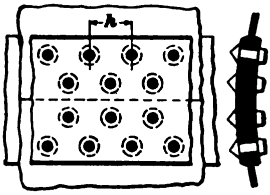
TYPE OF JOINT AND FORMULAS FOR STRENGTH

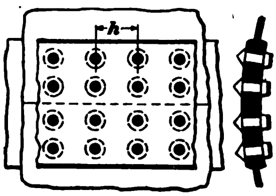
Double-riveted lap joint, with one cover-plate and staggered riveting.



$$R_t = (h - d)\,t\,S_t$$
$$R_s = 3.14\,d^2\,S_s$$
$$R_{st} = (h - 1.5d)\,t\,S_t + .7854\,d^2\,S_s$$
$$R_{cs} = 2.5\,d\,t\,S_c + 1.18\,d^2\,S_s$$
$$d = .64\,\frac{t\,S_t}{S_s}$$
$$h = \frac{3.14\,d^2\,S_s}{t\,S_t} + d$$

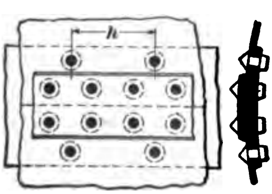
Double-riveted butt joint, with equal cover-plates, and staggered riveting or chain riveting.





$$R_t = (h - d)\,t\,S_t$$
$$R_s = 2.75\,d^2\,S_s$$
$$R_c = 2\,d\,t\,S_c$$
$$R_{st} = (h - d)\,t\,S_t + 1.38\,d^2\,S_s$$
$$R_{cs} = d\,t\,S_c + 1.38\,d^2\,S_s$$
$$d = .73\,\frac{t\,S_c}{S_s}$$
$$h = \frac{2.75\,d^2\,S_s}{t\,S_t} + d$$

Double-riveted butt joint, with unequal cover-plates, and chain riveting.

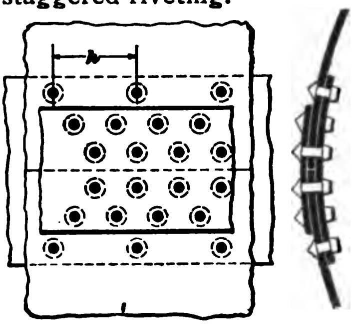


$$R_t = (h - d)\,t\,S_t$$
$$R_s = 3.53\,d^2\,S_s$$
$$R_c = 3\,d\,t\,S_c$$
$$R_{st} = (h - 2\,d)\,t\,S_t + .7854\,d^2\,S_s$$
$$R_{cs} = 2\,d\,t\,S_c + .7854\,d^2\,S_s$$
$$d = .85\,\frac{t\,S_c}{S_s}$$
$$h = \frac{3.53\,d^2\,S_s}{t\,S_t} + d$$

TABLE III—(*Continued*)

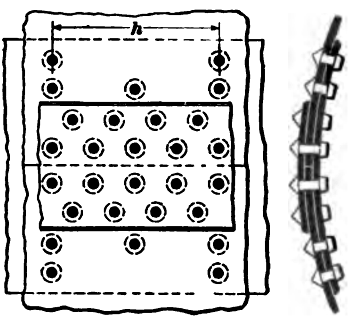
TYPE OF JOINT AND FORMULAS FOR STRENGTH

Triple-riveted butt joint, with unequal cover-plates and staggered riveting.



$$R_t = (h - d)\,t\,S_t$$
$$R_s = 6.28\,d^2\,S_s$$
$$R_{st} = (h - 2d)\,t\,S_t + .7854\,d^2\,S_s$$
$$d = 1.27\,\frac{t\,S_t}{S_s}$$
$$h = \frac{6.28\,d^2\,S_s}{t\,S_t} + d$$

Quadruple-riveted butt joint, with unequal cover-plates and staggered riveting. Thickness of straps in this particular joint should be at least three-fourths that of the plate.



$$R_t = (h - d)\,t\,S_t$$
$$R_s = 13.35\,d^2\,S_s$$
$$R_{st} = (h - 4d)\,t\,S_t + 2.36\,d^2\,S_s$$
$$d = 1.27\,\frac{t\,S_t}{S_s}$$
$$h = \frac{13.35\,d^2\,S_s}{t\,S_t} + d$$

REMARK.—The formulas of this table are based on the principles of Strength of Materials and Machine Design. Their derivations are given under Strength of Riveted Joints in the Section on Machine Design, in another volume, and are not repeated at this point. The formulas are again stated here for the purpose of having them convenient for reference in connection with the text on Steam Boiler Design, and also to keep them before the student during his study of that subject. They should be well understood, as they are applied in the design of almost all types of steam boilers.

This is somewhat greater than $\frac{5}{16}$ inch, the nearest commercial plate thickness, and as the boiler plate may be eventually weakened by corrosion, the plate thickness will be taken as $\frac{3}{8}$ inch.

To determine the diameter of rivet, use the formula for d for a double-riveted butt joint with unequal cover-plates as given in Table III. Then,

$$d = .85\frac{tS_c}{S_s} = \frac{.85 \times \frac{3}{8} \times 90{,}000}{44{,}000} = .652 \text{ inch}$$

Taking d equal to $\frac{5}{8}$ inch, which is less than .652 inch, in order that the resistance to crushing may be less than the resistance to shearing, the pitch of the rivets in the outer row may be found by the formula for h in Table III, thus:

$$h = \frac{3.53\,d^2S_s}{tS_t} + d = \frac{3.53 \times \frac{25}{64} \times 44{,}000}{\frac{3}{8} \times 53{,}000} + \frac{5}{8} = 3.678 \text{ inches}$$

In order that the resistance to tearing along the outer row may not become greater than the other resistances, take the pitch h as $3\frac{3}{8}$ inches, instead of $3\frac{5}{8}$ inches. Then, knowing the values of d and h, the values of R_t, R_s, R_c, R_{st}, and R_{cs} may be calculated by the formulas given in Table III, thus:

$$R_t = (h - d)\,t\,S_t = (3\tfrac{3}{8} - \tfrac{5}{8})\tfrac{3}{8} \times 53{,}000 = 54{,}656 \text{ pounds}$$
$$R_s = 3.53\,d^2S_s = 3.53(\tfrac{5}{8})^2 \times 44{,}000 = 60{,}672 \text{ pounds}$$
$$R_c = 3\,d\,t\,S_c = 3 \times \tfrac{5}{8} \times \tfrac{3}{8} \times 90{,}000 = 63{,}281 \text{ pounds}$$
$$R_{st} = (h - 2d)\,t\,S_t + .7854\,d^2S_s = (3\tfrac{3}{8} - \tfrac{10}{8})\tfrac{3}{8} \times 53{,}000 + .7854(\tfrac{5}{8})^2 \times 44{,}000 = 55{,}733 \text{ pounds}$$
$$R_{cs} = 2\,d\,t\,S_c + .7854\,d^2S_s = 2 \times \tfrac{5}{8} \times \tfrac{3}{8} \times 90{,}000 + .7854(\tfrac{5}{8})^2 \times 44{,}000 = 55{,}686 \text{ pounds}$$

Of the above values, R_t is the least, and it will therefore determine the efficiency of the joint. The strength of the solid plate for a width equal to h is $h\,t\,S_t = 3\frac{3}{8} \times \frac{3}{8} \times 53{,}000 = 67{,}078$ pounds. The efficiency of the joint is, therefore, $\frac{54{,}656}{67{,}078} = .815$, or 81.5 per cent. Or, using the formula for joint efficiency,

$$y = \frac{h - d}{h} = \frac{3\frac{3}{8} - \frac{5}{8}}{3\frac{3}{8}} = .815, \text{ or } 81.5 \text{ per cent.}$$

The efficiency of the girth seam does not need to be more than half that of the longitudinal seam. Using the same size of rivet but making the pitch h one-half as great as in the butt joint, or $1\frac{11}{16}$ inches, the efficiency of the single-riveted lap joint of the girth seam is

$$y = \frac{h-d}{h} = \frac{1\frac{11}{16} - \frac{5}{8}}{1\frac{11}{16}} = .63, \text{ or 63 per cent.,}$$

which is greater than half the efficiency of the longitudinal joint. Hence, a girth seam in which $d = \frac{5}{8}$ and $h = 1\frac{11}{16}$ is amply strong.

The actual factor of safety for the shell as a whole is affected by the efficiency of the riveted joint. The safe strength of the shell at the longitudinal joint for a width h is $(h-d)t\frac{S_t}{f}$, and for the two sides of the shell $2(h-d)t\frac{S_t}{f}$. This value must be equal to the internal pressure for the length h, which is hpD. Hence, $hpD = 2(h-d)t\frac{S_t}{f}$. Then,

$$f = \frac{2(h-d)tS_t}{hpD} = \frac{2(3\frac{3}{8} - \frac{5}{8})\frac{3}{8} \times 53{,}000}{3\frac{3}{8} \times 120 \times 48} = 5.62$$

as the factor of safety of the boiler.

If the joint is perfect as regards design and workmanship, the strengths at the various possible places of failure should be approximately equal, or, to be on the safe side, the values R_s and R_c should be greater than R_t as indicated by preceding calculations. To obviate failure of the cover-plates, they should be made at least five-eighths of the thickness of the shell plate. For the joint under consideration they would have to be at least $\frac{5}{8} \times \frac{3}{8} = \frac{15}{64}$, say $\frac{1}{4}$, inch thick. To make ample provision for calking, it would, however, probably be advisable to make the thickness $\frac{5}{16}$ inch.

STRENGTH OF STAYS AND SUPPORTS

STAYS FOR FLAT SURFACES

9. Strength of Unstayed Flat Surfaces.—The pressure, in pounds per square inch, that may be safely sustained by a flat circular plate, without stays, supported at the edges, such as the head of a boiler shell, is given by the formula

$$p = \frac{3t^2S_t}{2r^2f} \qquad (1)$$

in which p = pressure, in pounds per square inch;
t = thickness of plate, in inches;
S_t = ultimate tensile strength of the material;
r = radius of the plate, in inches;
f = factor of safety.

By solving formula **1** for t,

$$t = r\sqrt{\frac{2pf}{3S_t}} \qquad (2)$$

It was shown in Art. **3** that the thickness of the boiler shell should be determined by the formula

$$t = \frac{fpD}{2S_t y} \qquad (3)$$

Comparison of the results obtained by formulas **2** and **3** will show that, for the same conditions, the unstayed flat head of a boiler must be much thicker than the cylindrical shell.

EXAMPLE.—Find the necessary thickness of an unstayed head and that of the shell of a wrought-iron boiler 3 feet in diameter, under a steam pressure of 80 pounds per square inch. Assume a tensile strength of 48,000 pounds per square inch, with a factor of safety of 6 and an efficiency of the riveted joint of 60 per cent.

SOLUTION.—Substituting the given values in formula **2**,

$$t = r\sqrt{\frac{2pf}{3S_t}} = \frac{36}{2}\sqrt{\frac{2 \times 80 \times 6}{3 \times 48{,}000}} = 1.47, \text{ say } 1\tfrac{1}{2}, \text{ in.}$$

Using formula **3** for the thickness of the shell,

$$t = \frac{fpD}{2S_t y} = \frac{6 \times 80 \times 36}{2 \times 48{,}000 \times .60} = .3, \text{ say } \tfrac{5}{16}, \text{ in. Ans.}$$

Thus, the unstayed flat head would need to be almost five times as thick as the shell. It is apparent, therefore, that flat heads of moderate thicknesses must be well stayed.

10. When a square plate, fixed along the edges and unsupported by stays, is subjected to a uniformly distributed pressure, the thickness is found by the formula

$$t = \frac{a}{2}\sqrt{\frac{pf}{S_t}}$$

in which a is the length of one side of the plate and the other symbols have the same meanings as in Art. **9**. When the thickness required for strength, as found by the formula just given, becomes excessive, a plate of ordinary thickness, well stayed, is employed.

EXAMPLE.—Find the thickness of an unstayed firebox plate 40 inches square and exposed to a steam pressure of 100 pounds per square inch, assuming that $S_t = 48{,}000$ and $f = 6$.

SOLUTION.—Applying the above formula,

$$t = \frac{a}{2}\sqrt{\frac{p\,f}{S_t}} = \frac{40}{2}\sqrt{\frac{100 \times 6}{48{,}000}} = 2\tfrac{1}{4} \text{ in., nearly. Ans.}$$

Such a thickness is, of course, out of the question, and a thin plate well supported by stays would be used instead.

11. Maximum Stress on Flat Plates.—The rules issued by the Board of Supervising Inspectors of the Steamboat Inspection Service, Department of Commerce and Labor of the United States, limit the working pressure on flat stayed surfaces as follows: All stayed surfaces formed to a curve whose radius is over 21 inches, except surfaces otherwise provided for, are considered flat surfaces, and the maximum working pressure is determined by the formula,

$$p = \frac{c\,t^2}{a^2}$$

in which

p = working, or gauge, pressure, in pounds per square inch;

c = a constant having the following values: 112 for screw stays with riveted heads and plates $\frac{7}{16}$ inch thick, or less; 120 for screw stays with riveted heads and plates more than $\frac{7}{16}$ inch thick, or for screw stays with nuts and plates $\frac{7}{16}$ inch thick or less; 125 for screw stays with nuts and plates above $\frac{7}{16}$ inch and under $\frac{9}{16}$ inch thick; 135 for screw stays with nuts and plates $\frac{9}{16}$ inch thick or above; 170 for stays with double nuts, one nut on the inside and one on the outside of the plate, without washers or doubling plates; 160 for stays fitted with washers or doubling strips having at least half the thickness of the plate and a diameter of at least half the greatest pitch of the stays, riveted to the outside of the plates, and having one nut inside of the plate and one nut outside of the washer or doubling strip; t is then taken as 72 per cent. of the combined thickness of the plate and the washer or the plate and the doubling strip; 200 for stays fitted with doubling strips that have a thickness equal to at least half of the thickness of the plate reinforced and covering the full area braced (up to the curvature of the flange, if any), riveted to either the inside or outside of the plate, and stays having one nut outside and one inside of the plates; doubling plates must be substantially riveted; t is then taken at 72 per cent. of the combined thickness of the two plates; 200 for plates stiffened with tees or angle bars having a thickness of at least two-thirds the thickness of plate and depth of webs at least one-quarter the greatest pitch of the stays, and substantially riveted on the inside of the plates, and stays having one nut inside bearing on washers fitted to the edges of the webs that are at right angles to the plate; t is then taken as 72 per cent. of the combined thickness of web and plate;

t = thickness of plate, in sixteenths of an inch;

a = greatest pitch of stays, in inches.

The maximum pitch of stays, measured from center to center, must not exceed 18 inches.

EXAMPLE 1.—What is the allowable working pressure on a flat plate $\frac{7}{16}$ inch thick, fitted with screw stays having a pitch of 5 inches one way and 6 inches the other?

SOLUTION.—In this example, $c = 112$, $t = 7$, and $a = 6$. Then, applying the formula,

$$p = \frac{c\,t^2}{a^2} = \frac{112 \times 7^2}{6^2} = 152.4 \text{ lb. per sq. in. Ans.}$$

EXAMPLE 2.—What is the allowable working pressure on a flat plate $\frac{1}{2}$ inch thick, braced by stayrods spaced 12 inches from center to center and reinforced by washers 6 inches in diameter and $\frac{3}{8}$ inch thick, riveted to the plate, the stayrods having one nut inside the plate and one nut outside the washer?

SOLUTION.—In this example, $c = 160$ and $a = 12$. The value of t is 72 per cent. of the combined thickness of the plate and washer, that is, $.72\left(\frac{1}{2} + \frac{3}{8}\right) = .72 \times \frac{7}{8} = .63$ in., which, expressed in sixteenths, is $.63 \times 16 = 10.08$. Therefore, $t = 10.08$. Substituting in the formula,

$$p = \frac{c\,t^2}{a^2} = \frac{160 \times 10.08^2}{12^2} = 112.9 \text{ lb. per sq. in. Ans.}$$

12. Diameter of Direct Stays.—The diameter of a direct stay may be found as follows:

Let A = area, in square inches, of plate, supported by one stay;

d = smallest diameter of stay, in inches;

p = pressure of steam, in pounds per square inch;

T = safe tensile stress, in pounds per square inch, allowed in the stay. This is equal to the ultimate tensile strength divided by the factor of safety, or $\frac{S_t}{f}$.

The actual pressure on the stay is Ap and the safe load allowable on the stay is equal to its area multiplied by its safe tensile strength, or $.7854\ d^2 T$. Equating these two expressions and solving for the diameter of the stay,

$$.7854\ d^2 T = Ap,$$

or

$$d = 1.13\sqrt{\frac{Ap}{T}} \qquad (1)$$

Firebox staybolts are usually arranged so as to have the same pitch vertically and horizontally. Denoting this pitch by a, each staybolt must support an area a^2, which, substituted for A in formula **1**, gives

$$d = 1.13\sqrt{\frac{a^2 p}{T}} = 1.13\, a\sqrt{\frac{p}{T}} \qquad (2)$$

Solving formula **2** for a,

$$a = .885\, d\sqrt{\frac{T}{p}} \qquad (3)$$

The values of T to be used in the above formulas are as follows:

For copper screw staybolts, $T = 4{,}000$.

For iron screw stays and for other iron stays not exceeding $1\frac{1}{2}$ inches effective diameter, and for all stays that are welded, $T = 6{,}000$.

For unwelded iron stays above $1\frac{1}{2}$ inches effective diameter, $T = 7{,}500$.

For steel screw stays, and for other stays not exceeding $1\frac{1}{2}$ inches effective diameter, $T = 8{,}000$.

For steel stays above $1\frac{1}{2}$ inches effective diameter, $T = 9{,}000$.

It should be observed that no steel stays are to be welded.

EXAMPLE.—What is the effective diameter of the steel stayrods of a boiler, the rods being spaced 14 inches each way and the steam pressure being 135 pounds per square inch?

SOLUTION.—Using formula 2 and making $a = 14$, $p = 135$, and $T = 9{,}000$,

$$d = 1.13 \times 14\sqrt{\frac{135}{9{,}000}} = 1\tfrac{15}{16} \text{ in. Ans.}$$

13. Allowable Pressure on Direct Stays.—According to the rules prescribed by the Board of Supervising Inspectors of Steam Vessels, the maximum stress allowable, in pounds per square inch of cross-sectional area for stays used in the construction of marine boilers, when the same are accurately fitted and properly secured, shall be ascertained by the following formula:

$$p = \frac{m\,c}{n}$$

in which p = working pressure in boiler, in pounds per square inch;

m = least cross-sectional area of stay, in square inches;

c = a constant, having values of 9,000, 8,000, 7,000, or 6,000, according to conditions;

n = area of surface supported by one stay, in square inches.

The value of c for tested steel stays exceeding $2\frac{1}{2}$ inches in diameter is 9,000; and for tested steel stays more than $1\frac{1}{4}$ inches in diameter and not exceeding $2\frac{1}{2}$ inches, the value of c is 8,000, provided such stays are not forged or welded. The ends may be upset for threading, however, on condition that they are afterwards thoroughly annealed. The value of c is 7,000 for a tested brace whose cross-sectional area is not less than 1.227 square inches and not more than 5 square inches, provided such brace is prepared at one heat from a solid piece of plate, without welds. The value of c is 6,000 for all stays not otherwise provided for.

EXAMPLE.—Calculate the working pressure for a stay 1 inch in diameter when the distance from center to center of adjacent stays is 6 inches.

SOLUTION.—Use the above formula, making $c = 6{,}000$, $m = 1 \times 1 \times .7854 = .7854$ sq. in., and $n = 6 \times 6 = 36$ sq. in. Then,

$$p = \frac{mc}{n} = \frac{.7854 \times 6{,}000}{36} = 130.9 \text{ lb. per sq. in. Ans.}$$

14. Diagonal Stays.—In case a stay is diagonal, as shown in Fig. 2, the formula given in Art. **12** cannot be used to determine its size. For the same area A and steam

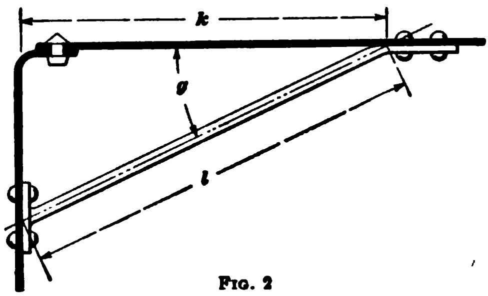


FIG. 2

pressure p, a diagonal stay must be larger than a direct stay. Let g represent the angle between the shell and the center line of the stay. Then, $\cos g = \frac{k}{l}$ and the force acting along the stay due to the pressure Ap is $\frac{Ap}{\cos g}$. Substituting this expression for the term Ap in formula **1**, Art. **12**, the diameter of a diagonal stay is found by the formula

$$d = 1.13\sqrt{\frac{Ap}{T \cos g}}$$

EXAMPLE.—Find the diameter of a diagonal stay that supports an area $6\frac{1}{2}$ inches by 8 inches against a steam pressure of 90 pounds per square inch, the angle between the stay and the shell being 25 degrees and the safe tensile stress being 7,000 pounds.

SOLUTION.—By the foregoing formula,

$$d = 1.13\sqrt{\frac{Ap}{T \cos g}} = 1.13\sqrt{\frac{6\frac{1}{2} \times 8 \times 90}{7{,}000 \times \cos 25°}}$$

$$= 1.13\sqrt{\frac{6\frac{1}{2} \times 8 \times 90}{7{,}000 \times .90631}} = .97, \text{ say 1, in. Ans.}$$

15. Girder Stays or Crown Bars.—The use of girder stays or crown bars to support the flat plates above the combustion chambers or fireboxes of boilers has already been mentioned in *Steam-Boiler Design*, Part 1. There seems to be, however, a wide variation in the methods and formulas by which the sizes of these stays or bars are calculated and the safe pressures allowed on them.

The Board of Supervising Inspectors of Steam Vessels recommends the following formula for finding the working pressure allowable on girders over combustion chambers:

$$p = \frac{c\,d^2\,t}{(w-a)\,h\,l}$$

in which p = working pressure allowable, in pounds per square inch;

c = a constant having the following values: 550 when the girder has one supporting bolt; 825 when the girder has two or three supporting bolts; 935 when the girder is fitted with four supporting bolts;

d = depth of girder, in inches;

t = thickness of girder, in inches;

w = width of combustion chamber, in inches;

a = pitch of supporting bolts, in inches;

h = distance from center to center of girders, in inches;

l = length of girder, in feet.

EXAMPLE.—Find the allowable working pressure in a boiler when the pitch of bolts along the girder is $7\frac{1}{2}$ inches, the distance between centers of girders is $7\frac{3}{4}$ inches, the length of girder is 2 feet $11\frac{1}{8}$ inches, the depth 8 inches, the thickness 2 inches, and the width of combustion chamber 34 inches, taking c = 825.

SOLUTION.—Substituting the given values in the above formula,

$$p = \frac{825 \times (8)^2 \times 2}{(34 - 7\frac{1}{2}) \times 7\frac{3}{4} \times 2.927} = 175.7 \text{ lb. per sq. in.} \quad \text{Ans.}$$

16. Assuming that a crown bar acts like a beam supported at the ends and uniformly loaded, its dimensions and the safe pressure it will sustain may be calculated by the formulas used for beams.

Let l = length of crown bar, in inches;
h = distance from center to center of crown bars, in inches;
p = steam pressure, in pounds per square inch;
b = breadth of crown bar, in inches;
d = depth of crown bar, in inches;
I = moment of inertia of the section;
c = distance of outermost fiber from the neutral axis;
W = total load on the crown bar;
f = factor of safety;
S_b = ultimate transverse strength of the crown bar, in pounds per square inch.

The bending moment on a simple beam uniformly loaded is $\frac{W l}{8}$. But $W = p h l$. Hence, the bending moment is $\frac{p h l^2}{8}$. The moment of resistance for a rectangular beam is $\frac{S_b I}{f c} = \frac{S_b b d^2}{6 f}$. Then, $\frac{p h l^2}{8} = \frac{S_b b d^2}{6 f}$. It is true that the crown sheet itself may aid in resisting the pressure, but it is not taken into consideration in the above discussion. The safe fiber stress $\frac{S_b}{f}$ in the girder or crown bar may be taken at from 8,100 to 9,000 for wrought iron, and at from 9,000 to 10,000 for steel. Solving the equation $\frac{p h l^2}{8} = \frac{S_b b d^2}{6 f}$ to find p,

$$p = \frac{8 S_b b d^2}{6 f h l^2} \qquad (1)$$

It is customary to make the breadth of the crown bar equal to one-fourth the depth. Substituting $\frac{1}{4} d$ for b in formula **1**,

$$p = \frac{S_b d^3}{3 f h l^2} \qquad (2)$$

To find the depth of a crown bar to sustain a given steam pressure, assuming that $b = \frac{1}{4} d$, solve formula **2** for d, and

$$d = 1.44 \sqrt[3]{\frac{f p h l^2}{S_b}} \qquad (3)$$

When high steam pressures are carried, it may be found impracticable to make simple girders strong enough. In such cases, radial or vertical stayrods are used to tie the girder to the outer shell of the boiler. The rule of the Board of Supervising Inspectors of Steam Vessels is that, when the steam pressure exceeds 160 pounds, girders must be suspended from the top of the shell by braces, each having a sectional area at least twice as great as the sectional area of each of the bolts suspending the crown sheet from the girder.

EXAMPLE 1.—Using the data of the example given in Art. **15**, find the safe working pressure by formula **2**, taking $\frac{S_b}{f} = 9{,}000$.

SOLUTION.—Substitute the given values in formula **2**. Then,

$$p = \frac{S_b d^2}{3 f h l^2} = \frac{9{,}000 \times 8^2}{3 \times 7\tfrac{3}{4} \times (35\tfrac{1}{8})^2} = 160.6 \text{ lb. per sq. in.} \quad \text{Ans.}$$

EXAMPLE 2.—The wrought-iron girder stays over a crown sheet are 30 inches long and spaced 9 inches between centers. The steam pressure being 150 pounds per square inch and the breadth of the girder one-fourth the depth, what is the necessary depth, assuming a safe fiber stress of 9,000 pounds per square inch?

SOLUTION.—Applying formula **3**,

$$d = 1.44\sqrt[3]{\frac{f p h l^2}{S_b}} = 1.44\sqrt[3]{\frac{150 \times 9 \times 30^2}{9{,}000}} = 7.387, \text{ say } 7\tfrac{3}{8}, \text{ in.} \quad \text{Ans.}$$

EXAMPLES FOR PRACTICE

1. What should be the diameter of firebox steel stay bolts spaced $4\frac{1}{2}$ inches apart, the steam pressure being 160 pounds per square inch? Ans. $\frac{23}{32}$ in.

2. Find the proper diameter of wrought-iron welded stayrods, pitched 12 inches apart each way and sustaining a steam pressure of 120 pounds per square inch. Ans. $1\frac{15}{16}$ in.

3. Determine the pitch of the screw staybolts of a locomotive firebox, the bolts being of wrought iron, $\frac{7}{8}$ inch in diameter, and the steam pressure 150 pounds per square inch. Ans. $4\frac{7}{8}$ in.

4. Find the diameter of an unwelded wrought-iron diagonal brace supporting an area of 45 square inches against a pressure of 80 pounds per square inch. The angle that the brace makes with the shell is 20 degrees. Ans. $\frac{15}{16}$ in., nearly

5. What pressure may be allowed on a plate $\frac{1}{2}$ inch thick stayed by staybolts having nuts inside and outside the plate, and pitched 8 inches between centers? Ans. 170 lb. per sq. in.

6. Find the allowable working pressure in a boiler when the pitch of the bolts along the girders is 8 inches, the pitch of girders 9 inches, the length of girders 3 feet, the depth 8 inches, the thickness 2 inches, and the width of the combustion chamber 35 inches, assuming that there are four supporting bolts to each girder.

Ans. 164.2 lb. per sq. in.

7. The crown sheet of a boiler is supported by steel girders 40 inches long and 8 inches between centers, the steam pressure being 125 pounds per square inch. Find the necessary depth of the girders, assuming that $b = \frac{1}{4}d$ and that $\frac{S_b}{f} = 10{,}000$. Ans. $7\frac{1}{16}$ in.

CORRUGATED FLUES

17. Thickness of Corrugated Flues.—The Board of Supervising Inspectors of Steam Vessels recommends the following formula for determining the safe working pressure in boilers having corrugated flues:

$$p = \frac{14{,}000\,t}{d} \qquad (1)$$

in which p = steam pressure, in pounds per square inch;
t = thickness of flue, in inches;
d = mean diameter of flue, in inches; or the inside diameter plus $1\frac{1}{2}$ inches plus the thickness t;
14,000 = a constant, to be used only when the corrugations are not less than $1\frac{1}{2}$ inches deep and not more than 8 inches from center to center.

By transforming formula **1**, an expression may be obtained by which to find the thickness of a corrugated flue when the diameter of the flue and the boiler pressure are known; thus:

$$t = \frac{p\,d}{14{,}000} \qquad (2)$$

EXAMPLE.—What should be the thickness of a corrugated furnace flue 42 inches in mean diameter, to carry safely a steam pressure of 150 pounds per square inch?

SOLUTION.—Applying formula **2**,

$$t = \frac{p\,d}{14{,}000} = \frac{150 \times 42}{14{,}000} = .45, \text{ say } \tfrac{1}{2}, \text{ inch. Ans.}$$

REINFORCING RINGS

18. Thickness and Width of Reinforcing Rings. The thickness of a single reinforcing ring may be from 1.25 to 1.5 times the thickness of the shell plate; when two rings are used, the thickness of each may be equal to the thickness of the shell. With one reinforcing ring the rivets are in single shear, while with two they are in double shear. Reinforcing rings must be securely riveted to the shell, and the diameter and number of rivets must be so proportioned that their combined resistance to shearing shall be at least equal to the resistance of the net section of the reinforcing rings in tension. The rings should be made of the same material as the shell. Whenever possible the short diameter of the manhole should lie lengthwise on the boiler.

Let w = width of reinforcing ring;
t = thickness of reinforcing ring;
d = diameter of rivet when driven;
t_1 = thickness of shell plate;
S_t = tensile strength of ring per square inch of section;
a = net section of the ring;
S_s = shearing strength of rivet per square inch of section;
l = length of opening in shell, lengthwise of the boiler;
n = number of rivets.

Then, the ring and the shell being of the same material, the strength of the part of the shell removed by cutting the hole is $l\,t_1\,S_t$, and the strength of the ring is $2\,a\,S_t$ for a single ring and $4\,a\,S_t$ for two rings. The net section for a single ring would be found by making $2\,a\,S_t = l\,t_1\,S_t$, or $a = \frac{l\,t_1}{2}$. The width of the ring is

$$w = \frac{a}{t} + d = \frac{l\,t_1}{2\,t} + d \qquad (1)$$

for a single row of rivets around the ring, or

$$w = \frac{a}{t} + 2\,d = \frac{l\,t_1}{2\,t} + 2\,d \qquad (2)$$

for two rows of rivets around the ring.

The net section of each ring where two rings are used is $a = \frac{l t_1}{4}$ and the width of each ring is

$$w = \frac{a}{t} + d = \frac{l t_1}{4 t} + d \qquad (3)$$

for a single row of rivets, and

$$w = \frac{a}{t} + 2d = \frac{l t_1}{4 t} + 2d \qquad (4)$$

for two rows of rivets around the rings.

A diameter of rivet should be assumed, for a trial, and the number of rivets calculated. If, in locating this number of rivets, the spacing or the number of rows becomes objectionable, a different diameter of rivet should be assumed and the number of rivets recalculated.

The trial nominal diameter of the rivets for a single reinforcing ring may be made about equal to the thickness of the shell plate plus $\frac{7}{16}$ inch; for two reinforcing rings it may be made about equal to the thickness of the shell plate plus $\frac{5}{16}$ inch.

The strength of a single ring in tension is $2 a S_t$, which should equal the shearing strength of all the rivets in one-half of the ring, or $\frac{n}{2} \times .7854 d^2 S_s$; that is,

$$2 a S_t = \frac{n}{2} \times .7854 d^2 S_s$$

Solving for the number of rivets in a single ring,

$$n = \frac{4 a S_t}{.7854 d^2 S_s} \qquad (5)$$

For two rings the strength of the rings in tension is $4 a S_t$, and the strength of the rivets in double shear in one-half of the ring is $\frac{1.75 n}{2} \times .7854 d^2 S_s$. Equating these values,

$$4 a S_t = \frac{1.75 n}{2} \times .7854 d^2 S_s$$

and solving for the number of rivets,

$$n = \frac{8 a S_t}{1.75 \times .7854 d^2 S_s} \qquad (6)$$

EXAMPLE 1.—A manhole opening is 11 inches by 15 inches, measuring 11 inches in the direction of the length of the boiler. If the shell plate is $\frac{1}{2}$ inch thick and a single-riveted reinforcing ring $\frac{5}{8}$ inch thick is to be used, how wide should it be? The rivets are to be 1 inch driven size.

SOLUTION.—Applying formula **1**,

$$w = \frac{11 \times \frac{1}{2}}{2 \times \frac{5}{8}} + 1 = 5.1, \text{ say } 5\tfrac{1}{8}, \text{ in. Ans.}$$

EXAMPLE 2.—How many rivets $\frac{13}{16}$ inch in diameter are to be used for a single reinforcing ring $\frac{1}{2}$ inch thick and 4 inches wide? Take the tensile strength of the ring as 60,000 pounds and the shearing strength of the rivets as 38,000 pounds per square inch of section. The reinforcing ring is to be single-riveted.

SOLUTION.—The driven size of the rivet is $\frac{13}{16} + \frac{1}{16} = \frac{7}{8}$ in. The net section of the ring is $(4 - \frac{7}{8}) \times \frac{1}{2} = 1.56$ sq. in. Applying formula **5**,

$$n = \frac{4 \times 1.56 \times 60{,}000}{.7854 \times (\frac{7}{8})^2 \times 38{,}000} = 17. \text{ Ans.}$$

EXAMPLE 3.—In a manhole reinforced by a pair of reinforcing rings, the rings are $\frac{3}{4}$ inch thick and $4\frac{1}{4}$ inches wide. With single riveting, how many $\frac{15}{16}$-inch rivets should be used? Take the tensile strength of the rings as 60,000 pounds per square inch and the shearing strength of the rivets as 38,000 pounds per square inch.

SOLUTION.—The driven size of the rivet is $\frac{15}{16} + \frac{1}{16} = 1$ in. The net section of the ring is $(4\frac{1}{4} - 1) \times \frac{3}{4} = 2.44$ sq. in., nearly. Applying formula **6**,

$$n = \frac{8 \times 2.44 \times 60{,}000}{1.75 \times .7854 \times 1^2 \times 38{,}000} = 23. \text{ Ans.}$$

BRACKETS

19. Shearing of Rivets in Brackets.—Boilers of the return-tubular type are commonly supported by cast-iron brackets riveted to the shell. These brackets are usually four in number, though six are often used, particularly if the boiler exceeds 16 feet in length. Each is generally riveted to the shell by from 7 to 9 rivets, the rivets being considerably larger and more widely spaced than those in the shell joints or seams. The brackets should be so placed that the flat under surfaces of the flanges are from $3\frac{1}{2}$ to $4\frac{1}{2}$ inches above the horizontal center line of the shell, and their distances from the ends of the boiler should be such as to divide the total load equally among them and cause the least possible bending moment on the shell. With four brackets, the distance of each from the nearest end of the boiler should be about one-fourth the length of the boiler.

Let W = total weight, in pounds, of boiler, water, fittings, etc.;
S_s = shearing strength of rivets in brackets, taken as 38,000 for wrought iron and 45,000 for steel;
d = diameter of rivets, in inches;
n = number of rivets in each bracket;
f = factor of safety.

Assuming that, through unequal settling of the brickwork, the total weight of the boiler might come on two brackets, each would carry a load of $\frac{1}{2}W$ pounds. The safe shearing strength of all the rivets in one bracket is $\frac{.7854\,d^2\,n\,S_s}{f}$. Then,

$$\frac{1}{2}W = \frac{.7854\,d^2\,n\,S_s}{f} \qquad (1)$$

Usually, the form of the bracket and the convenience of spacing determine the number of rivets, n, so that the diameter of rivets must be calculated. Solving formula **1** for d,

$$d = .798\sqrt{\frac{W f}{n\,S_s}} \qquad (2)$$

EXAMPLE.—The total weight of a boiler full of water is 30,000 pounds. Find the diameter of wrought-iron rivets, if there are 7 rivets in each bracket and the factor of safety is 6.

SOLUTION.—Apply formula **2**, making W = 30,000, f = 6, n = 7, and S_s = 38,000. Then,

$$d = .798\sqrt{\frac{30{,}000 \times 6}{7 \times 38{,}000}} = .656\text{, say } \tfrac{11}{16}\text{ in. Ans.}$$

Though $\frac{11}{16}$ in. is the calculated value, the rivets would doubtless be made larger, to give an excess of shearing resistance.

20. Tensile Strength of Rivets in Brackets.—The lowest row of rivets in a bracket is subjected to a tensile stress due to the tendency of the bracket to turn about its upper edge, and consequently these rivets should be calculated in tension. Thus, in Fig. 3, assume that half the total weight of the boiler acts at the end of the lug, as shown. The tendency is to turn the whole bracket about its upper edge *o*. Movement of this character is prevented by the lowest row of three rivets under the lug.

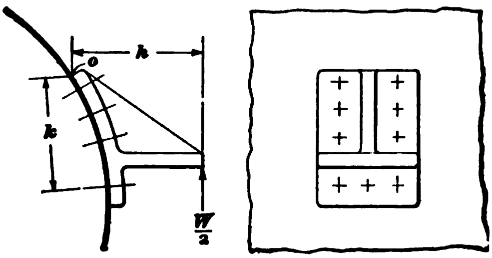


FIG. 3

Let W = total weight, in pounds, of boiler, water, fittings, etc.;
d = diameter of rivets, in inches;
c = number of rivets in lowest row;
S_t = tensile strength of rivets, taken as 48,000 for wrought iron and 55,000 for steel;
f = factor of safety, taken as 6;
h = moment arm of the force $\frac{1}{2}W$, in inches;
k = moment arm of the resistance of the lowest row of rivets, in inches.

The moment of the force $\frac{1}{2}W$ is $\frac{1}{2}Wh$, and the moment of resistance of the lowest row of rivets, in tension, taking the factor of safety into account, is $\frac{.7854\,d^2 c\,S_t\,k}{f}$. Equating these moments,

$$\frac{1}{2}Wh = \frac{.7854\,d^2 c\,S_t\,k}{f} \qquad (1)$$

and solving for d,

$$d = .798\sqrt{\frac{Whf}{ckS_t}} \qquad (2)$$

It may be that the diameter of rivets calculated by this formula will be greater than that required to resist shearing as found by formula **2**, Art. **19**. In any case, the greater diameter must be taken, since the rivets must be strong enough to withstand the greatest load put on them.

EXAMPLE.—If the total weight of a boiler is 30,000 pounds, and there are four brackets, in which h = 12 inches, k = 14 inches, and c = 3, what size of iron rivets should be used?

SOLUTION.—Substituting in formula **2**,

$$d = .798\sqrt{\frac{30{,}000 \times 12 \times 6}{3 \times 14 \times 48{,}000}} = .826, \text{ say } \tfrac{13}{16}, \text{ in. Ans.}$$

21. Strength of Brackets.—Brackets are frequently made of cast iron; as ordinarily made they will be strong enough to resist all stresses except, possibly, the bending stress. However, the strength in flexure may be determined

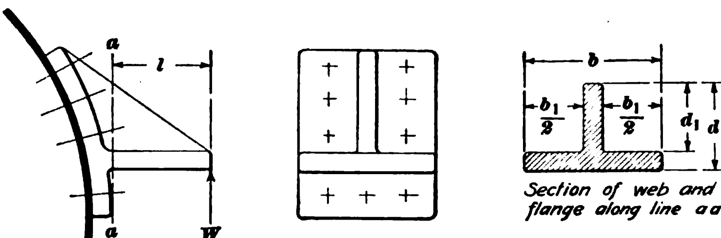


FIG. 4

by regarding the flange as a cantilever with the load at the end. Then, referring to Fig. 4, let

W = weight, in pounds, of boiler, water, fittings, etc.;
S_b = strength in flexure, taken as 30,000 for cast iron;
l = length of flange, in inches;
b = breadth of flange, in inches;
d = depth of web and flange, in inches, at section aa;
b_1 = breadth of flange minus thickness of web, in inches;
d_1 = depth of web, in inches, at section aa;
f = factor of safety, taken as 6.

Assuming the extreme condition, in which the whole load is carried by two brackets, the load on each is $\frac{1}{2}W$ pounds, and if this force acts at the outer end of the flange, the maximum bending moment is $\frac{1}{2}Wl$. The moment of resistance is $\frac{S_b I}{fc}$. But, for the **T**-shaped section at aa, the value of $\frac{I}{c}$ is

$$\frac{(b d^2 - b_1 d_1^2)^2 - 4 b d b_1 d_1 (d - d_1)^2}{6[d(bd - b_1 d_1) + b_1 d_1 (d - d_1)]}$$

Substituting this value for $\frac{I}{c}$ in the expression for the moment of resistance, and making the bending moment equal to the moment of resistance,

$$\tfrac{1}{2} W l = \frac{S_b}{f} \times \frac{(b d^2 - b_1 d_1^2)^2 - 4 b d b_1 d_1 (d - d_1)^2}{6[d(b d - b_1 d_1) + b_1 d_1 (d - d_1)]} \qquad (1)$$

Solving this equation for S_b,

$$S_b = \frac{3 W l f [d(b d - b_1 d_1) + b_1 d_1 (d - d_1)]}{(b d^2 - b_1 d_1^2)^2 - 4 b d b_1 d_1 (d - d_1)^2} \qquad (2)$$

Formula 2 is not adapted for designing the bracket, but by assuming the dimensions of the bracket and calculating the fiber stress it is possible to determine, by trial, a satisfactory size. The dimensions should be so chosen that the fiber stress will not be much in excess of 30,000 pounds per square inch.

EXAMPLE.—A boiler weighs 30,000 pounds when full of water, and is supported by cast-iron brackets. The length of the flange is 8 inches, the width 10 inches, the thickness $1\frac{1}{2}$ inches, and the web 8 inches deep and $1\frac{1}{8}$ inches thick. What is the fiber stress in flexure, if the factor of safety is 6?

SOLUTION.—Apply formula 2, making $W = 30{,}000$, $l = 8$, $f = 6$, $b = 10$, $d = 8 + 1\frac{1}{2} = 9\frac{1}{2}$, $b_1 = 10 - 1\frac{1}{8} = 8\frac{7}{8}$, and $d_1 = 8$. Then,

$$S_b = \frac{3 \times 30{,}000 \times 8 \times 6[9\frac{1}{2}(10 \times 9\frac{1}{2} - 8\frac{7}{8} \times 8) + 8\frac{7}{8} \times 8(9\frac{1}{2} - 8)]}{[10 \times (9\frac{1}{2})^2 - 8\frac{7}{8} \times 8^2]^2 - 4 \times 10 \times 9\frac{1}{2} \times 8\frac{7}{8} \times 8(9\frac{1}{2} - 8)^2}$$

$$= \frac{3 \times 30{,}000 \times 8 \times 6[9\frac{1}{2}(95 - 71) + 71 \times 1\frac{1}{2}]}{(902\frac{1}{2} - 568)^2 - 4 \times 95 \times 71(1\frac{1}{2})^2}$$

$= 28{,}231.6$ lb. per sq. in. Ans.

CALCULATIONS FOR A HORIZONTAL RETURN-TUBULAR BOILER

22. Requirements of Boiler.—It is required to design a 70-horsepower return-tubular boiler that is to use anthracite coal as fuel. The working steam pressure is to be 150 pounds per square inch gauge. The boiler is to evaporate 9 pounds of water from and at 212° F. per pound of coal, and 12 pounds of coal will be burned per square foot of grate area per hour. The heating surface should be at least thirty-six times the grate area. The tubes are to be made 15 feet long and 3 inches in diameter.

23. Grate Area.—In *Steam-Boiler Design*, Part 1, it was stated that the standard horsepower requires the evaporation of $34\frac{1}{2}$ pounds of water from and at 212° F. per hour for each horsepower. In the case under consideration, the water evaporated per hour would be $70 \times 34.5 = 2,415$ pounds. Then, from *Steam-Boiler Design*, Part 1, the grate area is found to be

$$G = \frac{W}{Fe} = \frac{2,415}{12 \times 9} = 22\tfrac{1}{3} \text{ square feet, nearly}$$

It is very common for the width of the grate to be made equal to the diameter of the boiler; hence, the determination of the dimensions of the grate can better be taken up after the diameter of the boiler has been decided on.

24. Number of Tubes.—The ratio of the tube area to the grate area may be 1 to 8. Then the total tube area $= 22\frac{1}{3} \div 8 = 2.8$ square feet, nearly. From the table on boiler tubes in *Steam-Boiler Design*, Part 1, the internal cross-sectional area of a 3-inch tube is 6.0787 square inches. The total number of tubes required is, therefore, $\frac{2.8 \times 144}{6.0787} = 66.3$, say 67, tubes. As the tubes are usually spaced alike on both sides of the vertical diameter of the boiler, this would necessitate sixty-eight 3-inch tubes.

25. Steam Space.—According to *Steam-Boiler Design*, Part 1, the steam space should be sufficient to contain the steam required for from 20 to 25 seconds. It should also be from one-fourth to one-third the cubic capacity of the boiler. The total weight of steam required per hour is equal to the water evaporated, or 2,415 pounds. For 25 seconds it would therefore be $\frac{2,415 \times 25}{60 \times 60} = 16.77$ pounds. From the Steam Table, 1 pound of steam at 150 pounds gauge pressure has a volume of about 2.756 cubic feet. The volume of 16.77 pounds would therefore be $16.77 \times 2.756 = 46.2$, say 46.5, cubic feet, nearly.

26. Diameter of the Boiler.—Taking the steam space as one-fourth the volume of the boiler (the tubes being for the present left out of the consideration) gives $46.5 \times 4 = 186$ cubic feet as the cubic contents of the boiler. As the boiler is to be 15 feet long, the area of cross-section of the steam and water space would be $186 \div 15 = 12.4$ square feet; to this must be added the tube area. The external cross-sectional area of a 3-inch boiler tube, from *Steam-Boiler Design*, Part 1, is 7.0686 square inches. The total area of cross-section of the tubes will be $\frac{7.0686 \times 68}{144} = 3.34$ square feet, nearly. Adding this to 12.4 gives 15.74 square feet as the area of cross-section of the boiler, and the diameter is $\sqrt{\frac{15.74}{.7854}} = 4.5$ feet, nearly. The diameter of the boiler (usually expressed in inches) is in this case 54 inches. If the width of the grate is made equal to the diameter of the boiler, the length of the grate will be the grate area divided by 4.5, or $22\frac{1}{3} \div 4.5 = 5$ feet, nearly.

27. Heating Surface.—The heating surface should now be compared with the grate area, to see that there is a ratio of 36 to 1. According to *Steam-Boiler Design*, Part 1, the heating surface of fire-tubes in boilers is the area of the internal surfaces of the tubes. To this is usually added half the area of the boiler shell and sometimes a portion of the boiler-head area; in this case, however, the head area will be neglected. From *Steam-Boiler Design*, Part 1, it will be found that 1.373 feet of 3-inch tube will give an internal heating surface of 1 square foot. For sixty-eight tubes each 15 feet long the heating surface will be $\frac{68 \times 15}{1.373} = 743$ square feet, nearly. The area of half of the shell will be

$$\frac{4.5 \times 3.1416 \times 15}{2} = 106 \text{ square feet, approximately}$$

Then, the total heating surface is $743 + 106 = 849$, which, divided by the grate area of $22\frac{1}{3}$ square feet, gives 38, nearly, as the ratio, which, although showing more heating surface than is commonly considered necessary, is very satisfactory. In case it is desired to reduce the heating surface, the number of tubes may be reduced to sixty-four, making a total heating surface of 805 square feet, with a ratio of 36, nearly.

28. Tube-Sheet.—The next step is to lay out the tube-sheet to see whether the water will cover the tubes to a sufficient depth and have sufficient evaporating surface for the steam to free itself from the water. Assuming that the steam space will occupy one-fourth the height of the boiler as a trial dimension, the highest water level *a b*, Fig. 5, is $\frac{54}{4} = 13\frac{1}{2}$ inches from the top of the boiler, and the length of the chord *a b* is $2\sqrt{(27)^2 - (13\frac{1}{2})^2}$ = 46.76, say $46\frac{3}{4}$, inches. In this particular case, the angle *a c b* is 120 degrees. For $\frac{gc}{bc} = \cos gcb = \frac{13.5}{27} = \frac{1}{2}$ and so gcb = 60 degrees; and as *a c b* is twice as great as *g c b*, it is 120 degrees. In any case, the angle may be determined by trigonometry, or even accurately enough by measuring with a protractor.

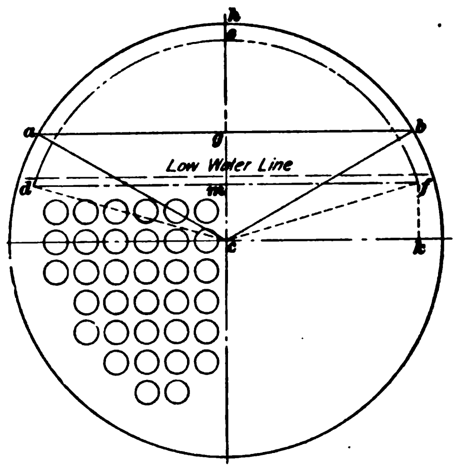


FIG. 5

Since acb = 120 degrees, the sector of the circle included by the radii *a c* and *b c* and the arc *a h b* is $\frac{120}{360}$ of the area of the whole circle, $\frac{120}{360} \times .7854 \times 4.5 \times 4.5 = 5.3$ square feet. The area of the triangle *a b c* is $ab \times \frac{1}{2} gc = 46.75 \times \frac{13.5}{2}$ = 315.56 square inches, or 2.19 square feet. Hence, the area *a b h* = 5.3 − 2.19 = 3.11 square feet, or the cross-section of the steam space. The steam space has been calculated as 46.5 cubic feet, and as the boiler is 15 feet long, 46.5 ÷ 15 = 3.1 square feet of cross-section is necessary.

Therefore, the area of 3.11 as found above is satisfactory. Should it be necessary to increase it an approximate method near enough for practical work is to divide the extra area required, in square inches, by the length of the chord *a b* and to subtract the quotient from the distance *c g*.

Allowing an extreme fluctuation of the water level of $5\frac{1}{2}$ inches, the low-water line would be $13\frac{1}{2} + 5\frac{1}{2} = 19$ inches below the top of the boiler or 8 inches above the horizontal diameter. It is common practice to assume the low-water level to be about 3 inches above the highest point of the crown sheet or tubes. If in this case 3 inches of water covers the tubes, the top row will have its center line $3\frac{1}{2}$ inches above the horizontal diameter. Then, by spacing the tubes $1\frac{1}{4}$ diameters, or $3\frac{3}{4}$ inches, apart, from center to center, both horizontally and vertically, and making the vertical center space $1\frac{1}{4}$ inches larger than the others, the tubes will be arranged as shown.

29. Thickness of the Boiler Shell.—In order to calculate the thickness of the shell, the factor of safety and the efficiency of the riveted joints should be known. By assuming a factor of safety of 5 and an efficiency of the riveted joint of 80 per cent., and calculating the thickness, an approximate value will be reached. Afterwards, the joint can be designed, the efficiency calculated, and then the factor of safety determined exactly. Applying the formula for thickness,

$$t = \frac{f p D}{2 S_t y} = \frac{5 \times 150 \times 54}{2 \times 60{,}000 \times .80} = \frac{27}{64} \text{ inch}$$

But, as $\frac{7}{16}$ inch is the nearest thickness of commercial boiler plate, that size will be taken.

From Table I, the rivet holes for $\frac{7}{16}$-inch plate and lap joint will have a diameter of $\frac{7}{8}$ inch and $2\frac{3}{16}$ inches pitch. By using a lap joint with an inside cover-plate, for the longitudinal seams, with $2\frac{3}{16}$ inches for the pitch of the inside rivets, the pitch of the outer rivets will be $4\frac{3}{8}$ inches, giving an efficiency of

$$y = \frac{h - d}{h} = \frac{4\frac{3}{8} - \frac{7}{8}}{4\frac{3}{8}} = .80, \text{ or } 80 \text{ per cent.}$$

The girth joints will be single-riveted, with an efficiency of

$$y = \frac{h - d}{h} = \frac{2\frac{3}{16} - \frac{7}{8}}{2\frac{3}{16}} = .60, \text{ or 60 per cent.}$$

The girth seam is evidently better able to resist rupture than the longitudinal seam. Hence, the latter should be calculated for strength against failure in all possible ways, as a check on the efficiency. This may be done by the use of the formulas for the joint as given in Table III, following the method employed in the case of the double-riveted butt joint in Art. 8.

30. Other Details to be Considered.—The lengths of the sections of the shell are limited by the size of the plate obtainable and the reach of the riveting machine. The sections are often made of the same length, but this is unnecessary. Three sections are made, with the middle one an inside section, so that the diameter of the two outside sections is twice the thickness of the shell larger than the middle one. This is the usual way of arranging the sections, as the fire coming in contact with the front riveted joint will injure the plates less than if the middle section were made the larger. Handholes can be fitted to the front and back heads below the tubes, and a manhole placed on the top of the middle section of the boiler. A nozzle should be attached to the front section, with a dry pipe running toward the rear end and supported at different points of its length. The openings for the manhole, steam pipe, and handholes should be reinforced with frames and nozzles (which are generally steel castings) or with reinforcing rings.

The area of the flue leading to the chimney and the area of the chimney itself should be at least equal to the total internal cross-sectional area of the tubes.

31. Staying the Flat Heads.—In staying the heads, steel stayrods can be designed to carry the whole pressure, and the extra strength of the channel bars and angle irons will give an extra margin of safety. Spacing the rods so as to distribute the total load as nearly equally over the ends as possible, the size of rods may be calculated.

Assuming that there are five stayrods, that the shell supports the head for 2 inches all around, and that the tubes support the heads for 2 inches above them, or 7 inches above the center line of the boiler, Fig. 5, the rods support the area enclosed by the arc def and the chord df. To find this area, it will first be necessary to determine the angle dcf, which is twice the angle ecf. The sine of the angle kcf is $\frac{fk}{cf} = \frac{7}{25} = .28$, which is the sine of $16\frac{1}{4}$ degrees, very nearly. The angle ecf = angle eck − angle $kcf = 90 - 16\frac{1}{4} = 73\frac{3}{4}$ degrees. Hence, the angle $dcf = 2 \times 73\frac{3}{4} = 147\frac{1}{2}$ degrees. The area of the sector $dcfe$ is, therefore, $\frac{147\frac{1}{2}}{360}$ of the area of a 50-inch circle, or $\frac{147\frac{1}{2}}{360} \times .7854 \times 50^2 = 804.5$ square inches. The area of the triangle dfc is $\frac{df \times mc}{2} = \frac{2mf \times mc}{2} = \frac{2ck \times mc}{2} = \frac{2\sqrt{\overline{cf}^2 - \overline{fk}^2} \times mc}{2}$ $= \frac{2\sqrt{625 - 49} \times 7}{2} = 168$ square inches. Then, area $defmd$ = area $dcfe$ − area $dfc = 804.5 - 168 = 636.5$ square inches.

The total load on the stays is 636.5×150 pounds, and the load on one stay is $\frac{636.5 \times 150}{5} = 19{,}095$ pounds. Substituting this value for Ap in formula **1** of Art. **12**, and making $T = 9{,}000$,

$$d = 1.13\sqrt{\frac{19{,}095}{9{,}000}} = 1.65, \text{ say } 1\tfrac{11}{16}, \text{ inches}$$

32. Brackets.—The boiler should be supported by four cast-iron brackets. In order to determine whether the brackets are strong enough, their probable strength should be computed in the three ways given in Arts. **19**, **20**, and **21**. In order that the strength of the brackets may be calculated, it is necessary to find the weight of the boiler full of water, which may be done by finding the volume of each part, multiplying it by the weight of a cubic unit of the material, and adding the several products, thus:

	Pounds
Weight of the shell and heads	4,600
Weight of the tubes	3,400
Weight of stays, etc.	1,000
Weight of nozzles, fittings, etc.	3,200
Weight of water when boiler is full	11,800
Total weight	24,000

Assuming that 7 rivets will be used in each bracket and that the total weight may come on two brackets, the factor of safety being 6, the diameter of rivets is found by applying formula **2** of Art. **19**,

$$d = .798\sqrt{\frac{W f}{n S_s}} = .798\sqrt{\frac{24{,}000 \times 6}{7 \times 38{,}000}} = .587 \text{ inch}$$

Next, calculate the rivets in tension, assuming that there are three rivets in the lower row and that $h = 14$, $k = 12$, and $f = 6$. Then, applying formula **2**, Art. **20**,

$$d = .798\sqrt{\frac{W h f}{c k S_t}} = .798\sqrt{\frac{24{,}000 \times 14 \times 6}{3 \times 12 \times 48{,}000}} = .8619 \text{ inch}$$

Therefore, $\frac{7}{8}$-inch rivets would be used.

The bracket should also be calculated for strength as a cantilever. Assuming that the width of the flange is 12 inches, the thickness of the flange $1\frac{1}{2}$ inches, the length of the flange 8 inches, the web $1\frac{1}{4}$ inches thick and 8 inches deep, the maximum fiber stress may be calculated by formula **2**, Art. **21**; thus:

$$S_t = \frac{3 W l f [d(b d - b_1 d_1) + b_1 d_1 (d - d_1)]}{(b d^2 - b_1 d_1^2)^2 - 4 b d b_1 d_1 (d - d_1)^2}$$

$$= \frac{3 \times 24{,}000 \times 8 \times 6 [9\tfrac{1}{2}(114 - 86) + 86(9\tfrac{1}{2} - 8)]}{[12(9\tfrac{1}{2})^2 - 10\tfrac{3}{4}(8)^2]^2 - 4 \times 12 \times 9\tfrac{1}{2} \times 10\tfrac{3}{4} \times 8(9\tfrac{1}{2} - 8)^2}$$

$= 20{,}138$ pounds per square inch.

This being much smaller than 30,000, the bracket is amply strong.

33. Steam Nozzle.—Assuming a velocity of flow of steam from the boiler of 1,000 feet per minute, so that the drop in pressure will be small, the size of the nozzle may readily be calculated. The total volume of steam generated in 25 seconds was found to be 46.5 cubic feet, or $\frac{46.5 \times 60}{25}$ $= 111.6$ cubic feet per minute.

With a velocity of 1,000 feet per minute, the area of the steam nozzle should be $\frac{111.6 \times 144}{1{,}000} = 16.07$ square inches. The diameter of the pipe should be $d = \sqrt{\frac{16.07}{.7854}} = 4.523$ inches, or a 5-inch pipe would be used.

BOILER SPECIFICATIONS

34. When a boiler is to be built to fulfil certain requirements, it is customary for the purchaser to furnish a set of specifications according to which the boiler must be constructed. Most boilers used in stationary practice are, however, built in standard sizes to fulfil ordinary working conditions, and in this case the builder furnishes specifications of his boilers to intending purchasers.

An example of the blank form of specifications for a return-tubular boiler, as furnished by the Hartford Steam Boiler Inspecting and Insurance Company, is here given.

SPECIFICATION

For________Inch Horizontal Tubular Steam Boiler, Prepared by the Hartford Steam Boiler Inspection and Insurance Company,

*For*____

TYPE AND GENERAL DIMENSIONS.— ________boiler is to conform to the following conditions and requirements: It is to be of the horizontal tubular type, set with________front, and all parts and pieces are to be designed accordingly________.

It is to be________feet________inches long, outside, and________inches in diameter, measured on the outside of the smallest ring of plates. Heads are to be________feet________inches apart outside.

MATERIALS: QUALITY, THICKNESSES, AND TESTS.—Shell plates are to be________of an inch thick on the edges, of open-hearth firebox steel, having a tensile strength of not less than________pounds, nor more than________pounds per square inch of section, and an elastic limit of not less than half the tensile strength, with not less than ________per cent. of ductility as indicated by contraction of area at point of fracture under test, and by an elongation of________per cent. in a length of________inches.

Heads are to be________of an inch thick, of best open-hearth flange steel. All plates, both of shell and heads, are to be plainly stamped with name of maker, brand, and tensile strength; brands so located that they may be seen on each plate after the boiler is finished.

Each shell plate is to bear a coupon which shall be sheared off, finished up, or tested by, or for, the maker of the boiler, at his expense. *Each coupon is to fulfil the foregoing requirements as to strength and ductility, and stand bending down double when cold, when red hot, and after being heated red hot and quenched in cold water, without signs of fracture.* There is not to be more than______per cent. of sulphur, nor more than______per cent. of phosphorus in the chemical composition of the plates and heads. *All plates failing to pass these tests will be rejected.* All tests and inspections of material may be made at the place of manufacture prior to shipment. Certified copies of report of tests must be sent to the Hartford Steam Boiler Inspection and Insurance Company, at Hartford, Conn.______

RIVETING.—The longitudinal seams are to be of the______riveted butt-joint type with double covering strips. *They are to be arranged to come well above the fire-line of the boiler, and break joints in the ______ring courses in the usual manner.* The plates are to be planed on the calking edges before rolling.

All dimensions and proportions are to be as shown on accompanying drawing No.______.

The girth seams are to be of the single-riveted lap-joint type; rivets to be of the same size as those in longitudinal seams, and pitched______inches apart from center to center.

The rivet holes are to be either drilled in place, or punched at least $\frac{1}{4}$ of an inch less than full size; if the latter method is used, the plates, after punching, are to be rolled and bolted together, and the rivet holes drilled in place $\frac{1}{16}$ of an inch larger than the diameter of the rivets. The plates are then to be disconnected. All burrs are to be removed from the edges of the holes. Should any holes be in the least out of true, they are to be brought in line with a reamer or drill; if a drift pin is used for this purpose, the boiler will be rejected.

All rivets are to be driven by hydraulic pressure, wherever possible, and allowed to cool and shrink under pressure. This pressure is to be sufficient to completely fill the rivet holes, producing a tight joint.

The rivets are to be of the finest quality of soft steel, having a shearing resistance of not less than______pounds, and a tensile strength of not less than______pounds per square inch of section, with an elastic limit of not less than half the tensile strength, and an elongation of not less than______per cent. in a length of______inches. The rivets are to have the same limits for sulphur and phosphorus as specified for the plates and heads.

BRACES.—There are to be______braces 1__inch__in diameter in the boiler,______above the tubes on front head, and______on rear head, of the crowfoot form, arranged as shown on drawing. None of them is to be less than 3 feet 6 inches long, and each is to be fastened to shell and heads by two___inch rivets at each end; or solid steel, diagonal braces of approved pattern, and of equal strength to the former, may be used. Care is to be exercised in setting them that they may bear uniform tension. Crowfoot braces may be flat in body, if of equal strength to those specified above.______

Braces Below Tubes.—There are to be______braces below the tubes in the boiler. ______of these are to be through braces extending from head to head. Each brace is to be 1___inch___in diameter with a fork formed on rear end and secured with a___inch turned bolt and nut to a______crowfoot securely riveted to rear head_________. The front end of the brace is to be upset to a diameter of______ inches, threaded, and secured to front head with a nut and washer on both the inside and outside of head.

The______remaining______braces are each to be 1___inch___in diameter, and secured to rear head in the same manner as the through braces______the front end of brace is to be extended forwards, fitted to side of shell, and riveted there with two______inch rivets. All to be substantially as shown on accompanying diagram of tube head No.______.

Tubes: Size, Number, and Arrangement.—There are to be ______best lap-welded or seamless-drawn tubes, free from all surface defects,___inches in diameter,___feet___inches long, and not less than standard thickness,___set in vertical and horizontal rows, with a clear space between them vertically___and horizontally, of___inch, except the central vertical space which is to be______inches, as shown on accompanying diagram of tube head No.______.

Holes for tubes are to be neatly chamfered off on the outside. Tubes to be set with a Dudgeon expander, and beaded down at each end. Tubes to be round and straight, properly annealed on the ends, and guaranteed to have been tested to at least 500 pounds internal pressure.

Manholes.—There are to be two manholes, one 11 in. × 15 in., with______pressed-steel frame, double-riveted to inside of shell on top, and one 10 in. × 15 in. flanged in front head below tubes, with suitable plates, yokes, and bolts, the proportions of the whole such as will make them as strong as any portion of the shell of like area.

Boiler Supports.—There are to be______cast-iron brackets, ______on each side, each 12 inches wide, with a projection of______ inches from the boiler, and of sufficient height for securely riveting in place; or pressed-steel brackets of equal strength may be used. Cast-iron wall plates,_____inches long,_____inches wide, and_____inches thick, are to be furnished for each_____bracket to rest upon, and three rollers, 1 inch in diameter and_____inches long, are to be furnished for all except front brackets to rest upon, to allow free expansion of the boiler.

NOZZLES.—There are to be______cast nozzles, made of gun iron or steel, one______inches internal diameter______for__steam-pipe connection______and one______inches internal diameter______for__ safety-valve connection, each accurately squared on top flange, and securely riveted to______boiler shell on top______. Forged- or pressed-steel pipe flanges may be used in place of nozzles. The flanges of the nozzles to correspond, in diameter and thickness, with standard extra-heavy pipe fittings.

SMOKE OPENING.—There is to be an opening______by______ inches cut out of front connection on top for attachment of uptake or flue.

FEEDPIPE.—There is to be a hole tapped in front head for a brass bushing,______inches above the top of upper row of tubes, and ______inches from center of boiler, on left-hand side, for__inch feedpipe connection. The bushing is to be not less than 2 inches long, to permit both the external and internal feedpipes to be screwed into it not less than______inch.

Also furnish and put in a______inch feedpipe extending from front head back to within 2 feet of rear head of boiler, thence across the boiler to near shell on right-hand side. On this end place an elbow with the outlet pointed down, as shown on drawings. Feedpipe is to be properly hung from the braces.

BLOW-OFF PIPE CONNECTION.—There is to be a______pressed-steel pipe flange,______inches in diameter and______inch thick, riveted to bottom of shell, near rear end, and tapped to receive a ______inch______blow-off pipe.

FUSIBLE PLUG.—There is to be a fusible plug in rear head, 2 inches above top of upper row of tubes.

FITTINGS.—There is to be furnished______safety valve,______ inches in diameter, one______inch______steam gauge, three______ inch gauge cocks, and one______inch gauge glass______inches long, all to be of approved pattern, and the necessary holes to be made for their proper connection. If combination water column is used, the steam and water connections between it and boiler must be made by pipes not less than $1\frac{1}{4}$ inches in diameter.

CASTINGS FOR SETTING.—There is to be furnished a substantial cast-iron front, with all necessary anchor bolts,______feet long, closely fitting front connection doors with suitable fastenings to prevent warping, closely fitting furnace doors with liner plates,______rear connection door 16 in. × 24 in., with liner plates, grate bars for grate, pattern to be selected by purchaser of boiler,______inches long by______ inches wide, with suitable bearer bars for same, arch bars for rear connection, and all buckstaves, with the necessary bolts or tie-rods, and all other castings, or ironwork of any description necessary for the proper construction and setting of the boiler complete.

In General.—The intent of the foregoing specification is to provide for material and workmanship of the best quality, and any details of equipment not mentioned in this specification nor shown on the drawings, but necessary for the proper completion of the boiler ready for operation, and to be hereafter contracted for, must be of equally good quality.

The size and description of parts are to conform substantially to the details of the accompanying plan, and the boiler, complete, is to be delivered at _______ and all the material and workmanship is to be subjected to the inspection and approval of the Hartford Steam Boiler Inspection and Insurance Company.

DETERIORATION AND INSPECTION

DETERIORATION

35. Kinds of Corrosion.—**Corrosion** may be defined as the eating away or wasting of the plates due to the chemical action of impure water. It is probably the most destructive of the various forces that tend to shorten the life of the boiler. Corrosion is of two forms—*internal* and *external.*

36. Internal Corrosion.—Internal corrosion may present itself as uniform corrosion, pitting or honeycombing, or

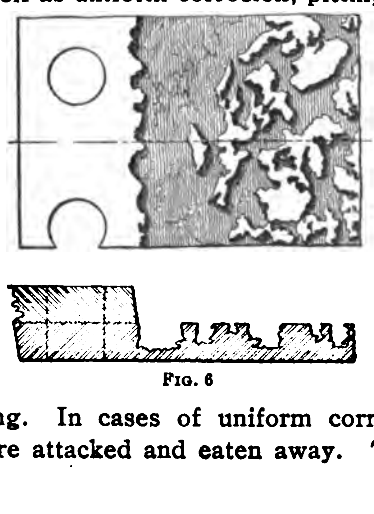


Fig. 6

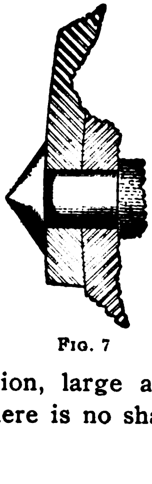


Fig. 7

grooving. In cases of uniform corrosion, large areas of plate are attacked and eaten away. There is no sharp line of division between the corroded part and the sound plate, and often the only way of detecting the corrosion is to drill a hole through the suspected plate and thus ascertain its thickness. Corrosion often violently attacks the staybolts and rivet heads.

Pitting and honeycombing are readily perceived. The plates are indented in spots with holes and cavities from $\frac{1}{32}$ to $\frac{1}{4}$ inch deep. The appearance of a pitted plate is shown in Fig. 6.

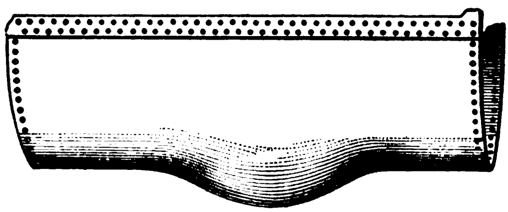
Grooving is generally caused by the buckling action of the plates when under pressure. Thus, the ordinary lap joint of a boiler distorts the shell slightly from a truly cylindrical form, and the steam pressure tends to bend the plates at the joint. This bending action is liable to start a small crack along the lap, which, being acted on by corrosive agents in the water, soon deepens into a groove, as shown in Fig. 7. The mark made along the seam by the sharp calking tool, when used by careless workmen, is almost certain to lead to grooving.

37. External Corrosion.—External corrosion frequently attacks stationary boilers, particularly those set in brickwork. The causes of external corrosion are dampness, exposure to weather, leakage from joints, moisture arising from the waste pipes, or blow-out. When leakage occurs in a joint that is hidden by the brickwork setting, the plates may be corroded very seriously without being discovered.

External corrosion should be prevented by keeping the boiler shell free from moisture, and by repairing all leaks as soon as they appear. Joints and seams should be in a position where they may be inspected for leaks.

Leakage at the seams may be caused by delivering the cold feedwater on to the hot plates; another cause is the practice of emptying the boiler when hot and then filling it with cold water. The leakage in both cases may be traced to the sudden contraction of the plates due to the sudden cooling. In any case, abrupt changes in the temperature of the shell should be avoided. The rush of cold air into the furnace of an externally fired boiler when the door is opened is a fruitful source of leakage and fracture. For this reason the shell should be constructed, if possible, so that none of the seams are in contact with the fire.

Overheating may be caused by low water or by incrustation. When the plate is covered with a heavy scale, the heat is not carried away by the water fast enough to prevent a rise of temperature, the plate becomes red hot and soft, and yields to the steam pressure, forming a pocket, as shown at



A

FIG. 8

A, Fig. 8. If the pocket is not discovered and repaired, it stretches until finally the material becomes too thin to withstand the steam pressure; the pocket bursts, and an explosion follows. The vegetable or animal oils carried into the boiler from the condenser are particularly liable to cause the formation of pockets.

INSPECTION AND TESTING

38. Hammer Test.—The condition of a boiler as regards safety can be determined only by careful inspection. Insured boilers are periodically inspected by experienced inspectors in the employ of the insurance company. The inspector notes the condition of the plates as to whether or not they are corroded or incrusted, and inspects the interior in search of broken stays or rivets or fractured joints. The condition of the plates is generally determined by tapping the plates with a light hammer; any weakness will immediately reveal itself to the skilled inspector, who is able to judge the thickness and soundness of the plate by the sound of the blow and the rebound of the hammer. This is known as the **hammer test.** When the thickness is a matter of doubt, a small hole may be drilled through the plate, and afterwards plugged up.

39. Hydrostatic Test.—Boilers subject to government inspection are submitted to the **hydrostatic test.** The boiler is filled with water, a pump is applied and more water is forced in, until the pressure exceeds that which the boilers are expected to carry. If the boiler stands the water pressure without fracturing or developing leaks, it is assumed that it will carry the required steam pressure in safety.

In making the hydrostatic test, the pressure must be applied very slowly and carefully, and the gauge watched for any drop of pressure that would denote a yielding of some part of the boiler. New boilers are tested by hydrostatic pressure to reveal leaky joints or rivets. When the seams or rivets are not tight, water trickles out in drops or spins out in a stream. Such places are marked with chalk and afterwards recalked. Boilers are usually tested hydrostatically to $1\frac{1}{2}$ times the pressure they are to carry.

The objection is raised against the hydrostatic test that there is danger of straining the plates beyond the elastic limits, and thereby a boiler may be permanently injured that would have been safe at the working steam pressure. The insurance companies in most cases depend on the hammer test, but use the hydrostatic test for new boilers, old boilers extensively repaired, and all boilers that cannot be examined thoroughly inside and outside.

A method of applying the hydrostatic test used by many engineers is to fill the boiler full of cold water and build a gentle fire in the furnace. As the temperature of the water rises it expands, and thus subjects the shell to pressure. It is urged in favor of this method that the pressure is raised steadily, and the boiler is not so liable to be injured as it is when subjected to sudden and jerky rises of pressure due to the working of a pump. The temperature of the water should in no case be allowed to rise above 212°, since, otherwise, if a rupture should take place, the pressure of the water would be lowered to that of the atmosphere, and, the temperature of the water being above the boiling point at atmospheric pressure, a quantity of the water might suddenly flash into steam and cause an explosion.

The inspection of steam boilers should begin at the place where the plates are manufactured, and continue as long as the boiler is in use.

BOILER EXPLOSIONS

40. Causes.—Boiler explosions are in nearly all cases due to one cause only—overpressure of steam. Either the boiler is not strong enough to carry its working pressure safely, or else the pressure has been allowed to rise above the usual point by the sticking or overloading of a safety valve or some similar cause.

A boiler may be unfit to bear its working pressure for any of the following reasons: (1) defective design; (2) defects in workmanship or material; (3) corrosion, and wear and tear in general; (4) mismanagement in operation.

41. Defective Design.—The following are faults in design that often lead to boiler explosions: The boiler is insufficiently stayed—the stays being too small or too few in number; the cutting away of the shell for the dome, manhole, and other mountings, without strengthening the edge of the plate around the hole; the boiler may be too rigidly fixed in its setting, and thus fracture on account of unequal expansion; defective water circulation in a boiler may lead to excessive incrustation and consequent explosion; a poorly designed feed apparatus, or safety valve, often leads to explosion.

42. Defects of Workmanship or Material.—Defects of workmanship and material may include the choice of faulty material, containing blisters, lamination, etc.; the careless punching and shearing of the plates; burned and broken rivets; plates burned, or otherwise injured in flanging, bending, or welding; scoring of the plates along the joints by sharp calking tools; injury of plates by the reckless use of the drift pin. Old plates may be injured, in patching them with new plates, by the operation of removing the old rivets and putting in the new ones, and also by the greater expansion and contraction of the new plate when exposed to fire.

43. Corrosion and Wear and Tear.—The strength of the shell may be weakened by corrosion, pitting, and grooving. In some boilers that have exploded, the plates have been found wasted to little more than the thickness of paper.

Fractures that ultimately end in explosion may be produced by letting the cold feedwater come directly in contact with the hot plates. It has already been remarked that the feed should be introduced into the coolest part of the boiler.

Vertical boilers hold the first place in the list of those liable to explosion. The ends of the tubes and the crown sheet are very liable to corrode; the crown sheet bulges downwards, and the reaction of the escaping steam may throw the boiler high in the air. Again, explosion may be the result of the collapse of the upper ends of the tubes—an accident that may occur when the tubes pass up through the steam space in vertical boilers.

It has been explained already how incrustation or the presence of grease may lead to the formation of pockets. If the pocket is not removed and replaced with a patch, rupture and explosion are liable to result.

44. Mismanagement.—The pressure of the steam may be allowed to rise above the normal blowing-off pressure by neglect or mismanagement. The safety valve may be neglected and allowed to stick fast to its seat. In one case, a safety valve recovered from a boiler explosion was found to be corroded to such an extent that a pressure of $1\frac{1}{8}$ tons was required to start it from its seat. Again, the safety valve may be temporarily shut off from the boiler by a stop-valve; numerous very disastrous explosions have been due to this cause. It cannot be emphasized too strongly that *a stop-valve should never be placed between the safety valve and* *the boiler.* Overpressure and consequent explosion may be caused also by the practice of overweighting the safety valve.

Low water is the cause ascribed to many explosions. It was formerly the custom to consider nearly every boiler explosion as due to shortness of water. It is now known, however, that externally fired boilers rarely explode on account of low water, though internally fired boilers may do so. In the case of boilers of the Lancashire type, a shortness of water leaves the furnace flue uncovered, when the flue becomes overheated and is liable to collapse; the same is true in regard to the fireboxes of the locomotive type. On the other hand, the tubes of a return-tubular boiler may stand a large amount of overheating before giving way, and it is next to impossible to burst a plain cylindrical boiler by low water, so long as it contains any water at all. Low water may lead to explosion, however, in the following manner: The uncovered plates become red hot, and, on being suddenly covered with fresh cold feedwater, may suddenly contract to such an extent as to produce rupture.

The superheating of the water has been supposed to have produced explosions in rare instances. It has been observed that water from which the air has been expelled may be heated 10° or 20° above its normal boiling point without any signs of boiling. On being agitated, as by the introduction of fresh feedwater, the volume of superheated water flashed suddenly into steam might, under favorable conditions, produce a pressure great enough to cause an explosion.

A weak boiler may possibly be exploded by the sudden opening or closing of a throttle valve. That such an explosion is possible is shown by the following experiment, made by United States Inspectors: A cylindrical boiler was tested and withstood a steam pressure of 300 pounds without injury. It was then filled again, and the steam pressure was run up gradually, the discharge valve being opened at intervals and the fall in pressure noted. When the valve was suddenly opened at a pressure of 235 pounds, the boiler was torn into fragments, the iron being twisted and torn and thrown in all directions. The sudden rush of steam from the boiler into the discharge pipe reduces the pressure in the boiler very rapidly; the reduction of pressure causes a sudden formation of a great quantity of steam within the water, and the heavy mass of water is thrown toward the opening with great violence, strikes the portions of the boiler near the opening, and breaks it open.

Explosions from this cause are probably rare; still, it is well to use caution in opening or closing stop-valves or safety valves.

45. Energy of an Explosion.—The destructive nature of a steam-boiler explosion is due to the enormous amount of energy stored in the steam and heated water. Professor Thurston calculated that a cubic foot of water heated at constant volume until its pressure is 60 to 70 pounds per square inch has about the same energy as a pound of gunpowder, and that the energy stored in a plain cylindrical boiler at 100 pounds pressure is sufficient to project it to a height of over 3½ miles.

If a boiler fractures while undergoing the hydrostatic test, the water escapes through the rent in the plate and no explosion takes place, because the cold water has little or no stored energy. But when a boiler filled with steam and water at high temperature fractures, a violent explosion generally follows. The steam escaping through the opening diminishes the pressure, and, consequently, a new body of steam is formed from the water, which, by escaping, lowers the pressure still more, allowing the formation of another new body of steam at a lower pressure, and this operation is continued until the pressure reaches that of the atmosphere. This formation of successive large bodies of steam, occurring, as it does, almost instantly, cannot fail to produce a disastrous explosion. Generally speaking, the larger the body of the contained water, the more disastrous is the result. The comparative safety of water tubes and sectional boilers is thus accounted for by the fact that they consist of numerous parts or sections, each containing a relatively small body of water. The bursting of one of these sections is unusual on account of its small diameter, and should it occur, no disastrous explosion would be likely to follow, on account of the small quantity of contained water.

Boiler explosions may be prevented by the use of properly designed and well-made boilers that have been correctly set and are under careful management. The boilers must be regularly inspected, repaired when necessary, and removed before becoming so worn out as to be dangerous.

Where boilers are to be placed in buildings, safety is the first object to be sought, and some form of sectional or water-tube boiler should be used. Where considerations of cost overrule those of safety, the tubular boiler is generally adopted.

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NOTE.—All items in this index refer first to the section (see the Preface), and then to the page of the section. Thus, "Absorption dynamometer, §40, p32," means that absorption dynamometer will be found on page 32 of section 40.

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